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No. 23

SHOCK AND VIBRATION

BULLETIN

FC

JUNE 1956

OFFICE OF
THE SECRETARY OF DEFENSE
Research and Development

Washington, D. C.

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CONFIDENTIAL **No. 23**

SHOCK AND VIBRATION

BULLETIN

JUNE 1956

OFFICE OF
THE SECRETARY OF DEFENSE

Research and Development

The 23rd Shock and Vibration Symposium was held in the Departmental Auditorium, Washington, D. C., on March 27-28, 1956. The Department of the Navy was host.

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FOREWORD

Laboratory Simulation was the central theme of the 23rd Shock and Vibration Symposium. The same general subject was discussed during the 11th Symposium in October 1948. In this interval of time considerable progress has been made technologically; and, the number of engineers and scientists who now take an active part in these meetings has grown significantly. The largest attendance in the entire series of symposia was recorded at the last gathering, and the volume of papers included in this Bulletin and in the separate unclassified mathematical Supplement represents the heaviest symposium publication we have ever issued. The content of these papers has also advanced in quality. Yet one wonders whether the 23rd Symposium provided a better and more effective interchange of needed information than was had six years ago.

Many of your comments and suggestions which have come in since the 23rd Symposium indicate that the technical worker did benefit from these meetings to a significant extent. Perhaps the importance and timeliness of the subjects discussed helped to increase the attendance of those who were searching for immediate knowledge of the field conditions in which their gear must operate. No doubt some were disappointed because, in several instances, there was not enough time for discussion and questions following each paper. These are some of the difficulties which beset the effort to disseminate technical information among a huge gathering. However, the editors believe that this large Bulletin and Supplement which contain all of the proceedings of the five sessions and which have been assembled in record time, will provide the reader promptly with much of the information he is seeking.

In the study of simulation it is of great importance to keep in mind just what field conditions a laboratory machine is expected to reproduce. Are we recreating the cause, the effect, or the damaging potential of shock and vibration? Another problem that is facing the shock and vibration engineer today is that of random vibration environment and its simulation in the laboratory. Should it be shown that random vibration is essential for adequate testing, a large and extremely expensive program of re-equipment will be necessary. However, if a satisfactory correlation can be shown to exist between the results of available sinusoidal test equipment and the statistical nature of random vibration, a tremendous potential saving to the defense effort will have been achieved. This subject is dealt with in a number of papers in the 23rd Bulletin, but a great deal of information and much more data are necessary before a decision can be reached.

Elias Klein

Any comments or requests pertaining to this Bulletin should be addressed to: Naval Research Laboratory, Code 5104, Washington 25, D. C.

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Jacobs, J. C.
Michalec, G. W.
Polster, W. J.
Ragusa, J. G.
Vavoudis, S. N.
Ward, J. C.

General Radio

Peterson, A. P. G.

General Tire & Rubber

Iredell, Robert
Miller, K. W.
Neher, E. P.
Tharp, T. A.

Glenn L. Martin

Barmat, Melvin
Englehardt, Richard
Evans, John
Forlifer, William
Hardesty, P. R.
Kelley, M. A.
Kirchman, Edward
Kirkley, E. L.
Luckock, W. J.
Schwab, Robert
Uebersax, Werner

Goodyear Aircraft

Hesler, W. H.
Rice, J. E.
Upp, R. L.

Grumman Aircraft

Pecan, Lester
Shergalis, Donald

Gulton Mfg.

Dranetz, A. I.
Orlacchio, A. W.

H. L. Yoh

Millard, E. J.

Hazeltine Electronics

Matson, D. L.

Holloman Air Dev. Ctr.

Heithecker, H. A.
Lange, Ernst
Sowell, E. N.

Houdaille Industries

Stickney, George

Hughes Aircraft

Curtis, A. J.
Diggle, F. V.
Felkel, E. O.
Glass, J. M.
Regal, W. J.
Schatz, G. W.
Schwartz, L. H.

Impact-O-Graph

Mielziner, W. S.

International R&D

Crawford, A. R.
Rushin, W. R.

Inter. Resistance

White, Sarah

Jack & Heintz

Caswell, P. M.
Thielman, R.

Jefferson Prov. Gd.

Everhart, R. A.
McKissick, F. A., CAPT

Jennings Radio

Daniels, J. W.

Jet Propulsion Lab.

Krause, Russell
Schimandle, William

Johns Hopkins Univ.

Ayre, R. S.

K. W. Johnson

Johnson, K. W.
Tobey, R. D.

Kimberly-Clark

Wamsley, D. C.

Kirtland AFB

Sanders, C. F.

Kollsman Instrument

Krause, J. K.

L.A.B. Corp.

Hill, Gilbert

L.O.F. Glass Fibers

Paul, H. W.

Lear Inc.

Altman, Otto
Monroe, J. P.

Lockheed Aircraft

Bumstead, C. R.
Della-Vedowa, R. P.
McGowan, N. E.
Mintz, Fred
Rice, S. M.
Williams, G. K.
Williams, R. M.

Lord Mfg.

Billman, G. H.
Bowler, J. A.
Goodill, J. J.
Ingoldsby, J. L.
Neely, R. S.
Painter, G. W.

Los Alamos Sci. Lab.

Russell, J. H.

M B Mfg.

Booth, G. B.
Cottle, H. N.
Dietrich, J. N.
Ellsworth, Warren
Jarowey, W. W.
Maki, Charles
Oravec, E. G.
Reen, G. K.
Unholtz, Karl

Magnavox

Juncal, R. W.

Marquardt Aircraft

Lonnquist, D. H.

McDonnell Aircraft

Lynch, F. R.
Merkel, R. W.
Meyer, D. D.
Scherrer, R. F.
Wimpey, J. L.

Michigan, Univ. of

Johnson, J. C.
Nourse, J. H.

Minneapolis-Honeywell

Bishop, E. L.
Chinquist, F. A.

NACA-Langley Field

Hubbard, H. H.
Youngblood, H. H.

NACA - Washington

Brown, H. H.

NAD - Earle

Kalista, George
Kelly, J. E.

NADC - Johnsville

Baker, C.
Blades, L. G.
Clark, L. P.
Cohen, A. A.

NADC - Johnsville (Cont'd)

Davidson, J. J.
Eiche, B.
Tait, J. N.

NAMC - Philadelphia

Kossayian, S. A.
Masi, Francis
Schulman, Marvin
Sharbaugh, Alvin
Siegel, Moses

NAMTC - Pt. Mugu

Kotonias, T. J.

NAS - Pt. Mugu

Holley, F. J.

NATC - Patuxent River

Bell, N. L.
James, Q. P.
Wilson, R. T.

National Union Elec.

Liddane, K. R.

Navy Subase - New London

Chapman, R. Y.
Engle, C. R., LT

Nav. Supply R&D Facility - Bayonne

Black, S.
Lajeniski, E.

NBS - Boulder Labs.

McClintock, R. M.

NBS - Washington

Bouche, R. R.
Edelman, Seymour
Jones, Earle
Lederer, P. S.
Marsden, C. P.
Ramberg, Walter
Rice, C. F.
Rouse, G. F.
Smith, E. R.
Wilson, B. L.

NEES - Annapolis

Bodnaruk, J.
Farmer, K. H.
Fotos, N. J.
Goodrich, D. G.
Shovestul, P. J.
Vallillo, J. M.
Walker, J. S.

NEL - San Diego

Carter, C. W.
Coleman, G. M.
Eitel, J. L.
Hauser, H. K.
Hudimac, A. A.

New York Air Brake

Edson, W. P.

New York Naval Shipyd

Berg, M. J.
Cohen, Levi
Knopfle, W. H.
Reitman, H. E.
Stickney, W. A.
Wohl, R. J.

NGF - Washington

Austin, P. H.
Busch, M. F.
Carpenter, S. O.
Duncan, R. C.
Hamill, T. E.
Lurie, W.
McKenzie, W. E.
Sanders, W. R.

NMDL - Panama City

Kelly, R. E.
Lively, G. P.
Lowry, R. C.
McKissock, G. D.
Ruthven, Beeman

No. American Aviation

Bailey, C. A.
Barber, W. J.
Franklin, P.E.
Hess, R. E.
Lunn, Rose E.
Marshall, A. H.
Rasmussen, N. A.
Reid, R. L.
Stewart, L. T.
Yurs, W. E.

NOEU - Washington

Allison, J. M.
Chaplick, R.
George, L. L.

NOL - Corona

Buus, M. L.
Fine, Aleck
Lambert, R. M.
Nestor, D. W.
Porter, D. W.
Whiteley, T. B.

NOL - White Oak

Armstrong, J. H.
Aucremanne, M. J.

NOL - White Oak

Baldwin, A. W.
Boeckel, J. H.
Brueggeman, W. C.
DeKnight, E. W.
Farley, F. W.
Fisher, J. C.
Fridinger, C. E.
Hanson, E. G.
Koch, H. S.
Kuczek, Rudolph
Logan, W. S.
Mead, R. F.
New, J. C.
Ploeger, T. R.
Reynolds, Delos
Rzepka, E. M.
Salmon, J. E.
Seely, C. R.
Shmueli, Kalman
Stathopoulos, G.
Stern, Daniel
Vagnoni, L. A.
Woolston, D. D.

NOP - Indianapolis

Heller, H. E.
Kuonen, C. E.
Welch, E. L.

Norfolk Naval Shipyd

Edwards, R. A.

Northrop Aircraft

Benner, R. C.
Bishop, N. S.
Cox, W. E.
Dillon, R. S.

NOTS - China Lake

Anderson, H. R.
Machowsky, John
McCullough, Foy

NOTS - Pasadena

Blatt, M. D.

NRL - Washington

Arnett, H. D.
Bachman, J. L.
Becke, Edward
Belsheim, R. O.
Blake, R. E.
Carson, R. H.
Conrad, R. W.
Cunningham, C. B.
Dick, A. F.
Forkois, H. M.
Gossett, J. D.
Halcombe, D. A.
Hanley, T. E.
Hardgrove, W. F.
Holcomb, W. L., Maj.
Hollings, A. J.

NRL - Washington (Cont'd)

Howard, A. L.
Kyser, R. H.
Krewson, W. L.
Lauver, R. E.
Marcus, Henri
Marzke, O. T.
Meyer, A. O.
Nowak, R. C.
O'Hara, G. J.
Oleson, M. W.
Pierson, Harry
Price, C. W.
Salzberg, B.
Seibert, E. R.
Shanahan, F. J.
Titus, J. W.
Trent, H. M.
Truitt, William
Vigness, Irwin
Wilson, B. J.
Wilson, W. O., LCDR
Worsley, D. A.

NUOS - Newport

King, R. T.
Murphy, F. G.

NUSL - New London

Burnham, B. B.
DeMoch, E. G.
Pierce, R. W.
Powell, J. G.

OASD (R&D)

Forsyth, P. S.
Furnas, C. C.
Mankey, W. A.
Randall, Henry
Swanson, A. A., Lt. Col.
Williams, E. B.

Off. Chief of Ord. - Army

Hansen, A. B.
Kaufman, Joseph

Off. Chief of Transportation - Army

Brown, E. H.
Stolarick, John

ONR

Bennett, Rawson, RADM
Crowley, J. M.
Liebowitz, Harold
MacCutcheon, E. M.
Tucker, J. F., CAPT

Ord. Ammunition Command - Joliet

Stirniman, J. P.

OSD

Bolles, R. B.

Owens-Corning Fiberglas

Westlake, J. M.

P. R. Mallory

Cook, W. O.

Penn State University

Brennan, J. N.
Eck, R. C.
Nitchie, F. R.

Philco

DeFrees, H. F.
Hewitt, L. H.
Sumerlin, W. T.

Phillips Petroleum

Bubb, F. W.
Stegelman, A. F.

Picatinny Arsenal

Leonardi, R. G.
Vecchio, R. A.

Pneumafil

Barr, J. W.
Deyton, J. B.
Sanborn, W. F.

Portsmouth Naval Shipyd

Galle, H. G.
Taylor, E. C.
Varrell, M. W.

Pratt & Whitney Aircraft

Gorton, R. E.
Krieghoff, R. T.

Puget Sound Naval Shipyd

Miles, W. H.
Watson, J. M.

Purdue University

Quinn, B. E.

QM Food & Container Inst.

Myers, E. C.
Williams, E. F.

QM R&D Center

Philleo, C. H.

RCA

Bantle, David
Cederbaum, Lawrence
DiTaranto, Rocco
Fuhrmeister, F.

RCA (Cont'd)

Hall, Wayne
Hawley, M. E.
Isom, W. R.
King, G. W.
Kressin, H. L.
Lohr, M. L.
Meyer, E. J.
Morris, C. N.
Newell, R. A.
Osgood, C. O.
Reeves, T. C.
Rich, Samuel
Schnapf, Abraham
Stadnyk, P.
Swartz, L.
Tabur, J.
Zartman, N. B.

Ramo-Wooldridge

Morrow, C. T.
Willett, B. M.

Raytheon Mfg.

Ball, E. W.
Batchelder, Laurence
Bowker, G. E.
Gladstone, Samuel
Keeler, P. R.
Roberts, P. V.
Sifferlen, J. J.
Thoms, D. B.
Wood, Ross
Wuth, H. M.

Redstone Arsenal

Allen, W. R.
Fritz, H. W.
Henderson, G. A.
Hunt, R. M.
Kline, L. V.
Koehn, D. C.
Lusser, Robert
Owens, L. J.
Pennington, William
Shipp, K. C.
Sullivan, H. D.
Taylor, J. M.
Watts, R. P.

Reed Research

Hobbs, E. V.

Remington Rand Univac

Einfeldt, R. B.
Hanson, G. S.
Harrison, W. F.
Haselberger, R. E.
Koenig, A. H.
Saltvick, A. C.

Rensselaer Polytechnic Institute

Macduff, J. N.

Republic Aviation

Caldwell, William
Ham, E. H.

Robinson Aviation

Nietsch, H. E.
Robinson, C. S.

Rock Island Arsenal

Hanson, J. C.
Smith, Arthur

Rocketdyne

Burowick, E. A.
Smith, D. M.

Rome ADC

Carroll, J. F.
Catenaro, E. A.
Darby, R. J.
Nares, H. E.

Rossford Ord. Depot

Borkenhagen, E. H.

Ryan Aeronautical

Anderson, L. E.
Puterbaugh, H. J.

S. S. Lee Assoc.

Lee, S. S.

San Fran. Naval Shipyd

Schrader, C. G.

Sandia

Ellett, D. M.
Othmer, R. T.
Westgate, H. E.
Williams, Don
Yorgiadis, A. J.

SCEL - Ft. Monmouth

Biamonte, O. A.
Devreotes, Peter
Kennedy, P. J.
Mattes, A. J.
Oliveri, J. J.
Trudell, B. J.
Umstead, H. C.
Webber, H. K.
Wilson, T. R.
Woolley, R. F.

SCSA - Philadelphia

Cawley, E. H.

Shock & Vibration Res.

McKay, R. L.

Sikorsky Aircraft

Smollen, Leonard

Sonotone

Bilsky, Herbert
Klingener, Harry
Myers, Henry

Southwest Res. Inst.

Sharp, J. M.

Sperry Gyroscope

Anderson, R. G.
Brenner, Harold
Bruno, E.
Flynn, Joseph
Frohrib, Darrel
Gulick, H. W.
Hawkins, Robert
Lord, Donald
Muller, J. T.
Pinto, S.
Steiner, Harry
Winter, D. I.

Springfield Armory

Julian, C. N.

Stanford Res. Inst.

Burgess, J. C.
Coale, C. W.

Sylvania

Lambert, Frank
Primpas, Louis
Robbins, J. D.
Rosato, Frank
Silva, James

Tinker AFB

Spillman, E. R.

U. S. Testing

Davidson, L. M.

United Aircraft

Tedder, J. A.

Vickers

Billet, A. B.

Vitro

Dorr, G. W.

WADC

Geist, R. E.
Golueke, C. A.
Granick, Neal

WADC - (Cont'd)

Grimm, J. R.
Hankey, R. K.
Kempton, N. P.
Kennard, D. C.
Magrath, H. A.
McCormick, J. W.
McDowell, Charles
McIntosh, V. C.
Paddock, G. B.
Rogers, O. R.
Sandoz, Luis
Sevy, R.
Wernicke, B. K.
Whitlock, R. S.

Walter Dorwin Teague Associates

Teague, W. D.

Watertown Arsenal

Baratta, F. I.
Bluhm, J. J.
Darcy, George
Landry, A. F.

Western Electric

Applegate, I. D.

Westinghouse

Allen, J. E.
Bruns, E. J.
Cartin, T.
Durkin, W. J.
Gilley, P. N.

Westinghouse (Cont'd)

Morris, H. R.
Reich, W. J.
Renshaw, J. H.
Sando, Robert
Wagner, R. J.
Wolfe, C. E.

Weston Elec. Instrument

Skidmore, W. H.

White Corp.

Hafkemeyer, E. E.

Wm. Miller Instruments

Hoskins, E. E.
Korb, Fred

Wright Engineering

Bradley, Wilson

WSPG

Bush, E. D.
Kinney, B. W.
McCabe, A. P.
Sheets, G. F.

Yale University

McKeehan, L. W.

WELCOMING ADDRESS

RADM Rawson Bennett, USN, Chief of Naval Research, ONR

It is a distinct pleasure for me to welcome this group of scientists and engineers to the 23rd Shock and Vibration Symposium. I am privileged to extend this welcome on behalf of your host, the Navy, and on behalf of the sponsor of these meetings, the Department of Defense. Since their beginning in 1947, these symposia have grown in size and in their importance to the national defense. The Navy is proud to have had a hand in the progress which has been made in this field.

These meetings are held to discuss ways and means toward effective protection of military equipments in service. As the complexity and cost of weapons increase, reliable performance becomes of vital importance and every factor which limits reliability in the field must be examined and corrected. The destructive forces of shock and vibration are a serious problem; and, it is to this group that the various organizations in the Department of Defense look to devise and develop effective countermeasures. This responsibility falls upon your shoulders, and it is my earnest hope and wish that you succeed in your undertaking.

We hear much talk today about the shortage of scientific manpower. It is said there are not enough engineers and physicists to attack the technical problems involved in national defense and civilian production. Moreover, the pace of technology is such that it is difficult for any group or individual worker to keep up with new developments in any particular field. That is why the government organizes these meetings; where recent advances in many areas of interest are made available to the defense technologist and current ideas are exchanged. In this way much unnecessary duplication of effort is avoided.

I believe it also is imperative in such a large and diversified group to promote a measure of education and training. The Department of Defense has a great stake in the availability of trained manpower, especially in a field like shock and vibration where engineering schools have only recently begun to include these subjects in their curricula. It is necessary for the better understanding of military needs and for the professional development of young engineers to introduce and elaborate here certain phases of this subject.

To sum up, the purpose of these meetings is the fruitful exchange of ideas and techniques to help you all to keep abreast in your military projects with the latest developments in the field. Because of the complexity of the problems which you face, some training and indoctrination of the young engineer has become necessary. We rely upon activities such as these to strengthen our country and maintain a high degree of readiness for any emergency.

I wish you every success.

OPENING REMARKS

Dr. C. C. Furnas
Assistant Secretary (Research and Development)
Department of Defense

PHILOSOPHY OF SIMULATION

I am very glad to have this opportunity of addressing the opening session of the 23rd Shock and Vibration Symposium.

These meetings which bring together research and development people from government laboratories, industry, and the universities, serve a really useful purpose in the dissemination of information leading to the improvement of military equipment in service.

The central theme of this Symposium, Laboratory Simulation, is a subject with certain aspects of which it has been my good fortune to be associated in the past. Before you plunge into technical details I would like to say a few

words on the philosophy of simulation and on the importance of simulation testing to the defense program.

First of all, what is laboratory simulation? Whenever you build a mockup in the laboratory to recreate, imitate, or reproduce conditions outside the laboratory you are simulating something. Essentially you are making believe that a real situation is in front of you which requires study. With this mockup you are attempting to approach reality as closely as possible so as to help you understand the actual problem. In simulating the shock and vibration experienced by an equipment or structure in the field, you are seeking to reproduce in the laboratory the damaging forces as they affect the proper functioning of the equipment.

Now why is laboratory simulation necessary? Why cannot the shock and vibration effects upon the mobile equipment be studied and neutralized in the field where the environmental conditions exist? The answers to these questions are manifold. Sometimes full-scale tests are not possible; very often they are too costly in money and in time, but a major drawback is that these tests produce such complex results that few can interpret them usefully. Because a complete understanding of the response of a structure to shock and vibration in the field is still lacking, it is necessary to make intelligent guesses and to attempt to distinguish the primary causes of damage from the mass of complex information available. These compromises and first approaches to solutions can best be made under controlled conditions in the laboratory; and, in the laboratory we find a wide variety of machines which have been devised to torture the equipment so as to reproduce part, if not all, of the environmental conditions of the field.

I would like to illustrate this need for laboratory simulation with two examples. The first example is from an actual project with which I was associated. Sometime ago a number of studies were undertaken to prevent injuries to pilots in airplane crashes. A particular investigation dealt with the problem of how heavy a bump a pilot's head could tolerate and what could be done about protection.

The most realistic test would have been to bump a human skull against the instrument panel or bulkhead as in a crash landing. Another method, and one which was actually tried in a few cases, subjected a human head to blows from a 10-lb pendulum bob. Successively greater heights were used until the subject yelled "Enough"! Such methods cause headaches and represent a lack of scientific objectivity.

Therefore the investigation was conducted in the laboratory on simulated heads made of various materials, and the results were much more definite. On these heads the impact necessary to crack the skull was quite reproducible for each material, and the helmet design evolved from this work proved effective on live heads. The results of these experiments showed the upper limits of acceleration which the human head can tolerate.

My second example is taken from an address given by Werner Von Braun before a meeting of the 14th Shock and Vibration Symposium in 1949. He related that during the development of the V2 rocket, at the time when the first service and troop training firings were conducted, it was found that a considerable percentage of the missiles disintegrated shortly before impact at an altitude of about one mile. After uncounted misinterpretations, some wind tunnel and other laboratory tests finally showed the cause of the trouble to be the fluttering of several skin panels, about one-third of the length from the tip. Thus it cost about 70 V2 in-flight trials to trace down this one source of trouble! A few simple laboratory tests would have avoided these terrific losses.

I could cite a wide variety of other examples to illustrate the value of laboratory simulation—some you will discuss during this Symposium. They will all demonstrate the ability of simulation testing to provide design information far more readily than it can be obtained from full-scale testing in the field.

The importance of simulation testing to the defense program stems directly from its economic and technical advantages over the full-scale test. The success of modern warfare depends largely on vehicle movement and vehicle speed, but as the speed increases the equipment carried is subjected to an increasingly severe shock and vibration environment. This in turn requires increasingly complex corrective measures to prevent malfunction. Today as never before trouble-free equipment is essential, and the performance and complete reliability of any system, man or machine, can be vital to the outcome of an engagement.

It is the designer's job to ensure the reliability of his equipment. Clearly he should know and understand the environmental conditions under which the gear he designs has to operate. Yet as we have seen, not only is the collection of such data difficult, but its interpretation is often impossible. Therefore, instead of beginning with the complex environmental conditions and working toward the simple in order to deduce the answers, we turn to laboratory simulation, and starting from the controlled, simple, and well understood, we build upwards from a firm foundation to the solution of our complex problems.

Someone has said, "Men often applaud an imitation and hiss the real thing." In environmental field conditions reality seems too complex for us to appreciate fully. Therefore, we not only applaud a good, simple imitation, we strive to achieve a meaningful make-believe.

PART I

GENERAL

CONCEPTS AND TRENDS IN SIMULATION

Charles E. Crede, Barry Controls, Inc.

Vibration testing procedures have underdone a pronounced evolution in recent years. This paper presents a review of vibration testing procedures leading to the present concept of the continuous spectrum or random excitation. Means used to define the vibration existing in an environment are discussed, and there is a comparison of scanning at discrete frequencies versus testing with random excitation.

The definition of simulate given by Webster is "to assume the mere appearance of, without the reality." When we speak of simulating an environment, we are not necessarily speaking of the creation of a condition having the same appearance as the environment. The principal purpose of the simulation is to establish a test adapted to reject equipment which is not qualified and to accept equipment which is qualified. When the environment being simulated is characterized by vibration conditions, the equipment under test is attached to a vibration testing machine whose motion is related to that of the airframe or other structure commonly used to support the equipment. Hopefully, the vibration test causes damage to unqualified equipment which is similar to that experienced in service, and appropriate design changes are made.

Vibration testing procedures used to qualify equipment for airborne service have undergone a pronounced evolution in a relatively short interval of time. Not many years ago, the conventional vibration test embodied a frequency range from 10 to 55 cps. One may suspect that the establishment of this frequency range was an expedient to make use of existing vibration testing machines. The testing machines which were used in that era were principally mechanical in nature. One of the more common types was a positive-drive machine driven by a crank and connecting rod or by a scotch yoke, whereas the other principal type embodied a flexibly supported table carrying an unbalanced flywheel.

This era of low-frequency tests ended rather abruptly when a data collection project revealed the existence of vibration embodying substantially higher frequencies. These frequencies and their associated amplitudes were plotted relative to amplitude-frequency coordinate axes, and an envelope was constructed around the plotted points. A vibration test specification conforming substantially to this envelope has become well established. This does not simulate any particular environment, but rather specifies a vibration condition which may be considered equal to the most severe condition reasonably expected to be encountered in service.

With the adoption of testing specifications requiring vibration tests at high frequency, the usefulness of the mechanical vibration machines began to decline. They are being supplemented by electrodynamic machines. Such a machine is essentially a large loud speaker having a dc field and an armature energized by alternating current whose frequency can be varied in conformance with the frequency of the desired vibration. The power source for the armature may be either an audio oscillator in combination with a large power amplifier or a bank of generators which may be driven at variable speeds. To date, the latter type of power source has seen substantially greater use. As the weight of the equipment to be tested and the required acceleration amplitude continue to increase, the armature becomes increasingly heavy and the force used in driving it increases.

With this vicious circle, the size and cost of testing machines tend to increase rapidly relative to the magnitude of useful testing force derived therefrom.

Considerable thought has been given to the rational specification of vibration tests, starting with the plot of amplitude versus frequency mentioned previously. Evidence indicates that damage occurs during a vibration test primarily as a result of a resonant condition. This suggests that the laboratory test embody the same frequencies that exist in the environment being simulated. A principal question then concerns the designation of an appropriate amplitude and duration of the test. Assuming that an amplitude which is representative of service conditions can be established, there is reason to suggest that the vibration test is completely valid only if it continues for a period equal to the flight life of the aircraft.

Insofar as piloted aircraft are concerned, such a requirement is completely impractical. A major problem then exists in determining how to decrease the duration of the vibration test. If the shorter test time is to be compensated for by an exaggerated or increased test amplitude, it is difficult to determine the law which governs the required degree of amplitude increase. Considering only structural failure, use may be made of the conventional stress versus cycles-to-failure relation developed during fatigue or endurance tests of metals. It is necessary to take care in selecting the degree of increase, however, to avoid nonlinearity of the structure under test. Furthermore, it is not known whether damage to and maloperation of equipment and its components, such as vacuum tubes and relays, are covered by the same laws as fatigue of structural material.

In establishing vibration testing programs, cognizance should be taken of the significantly different requirements of piloted aircraft and guided missiles. One important distinction is the duration of flight. The expected flight time of a piloted aircraft is invariably much greater than any reasonable test period; on the other hand, the flight time of most guided missiles is relatively short. It is quite reasonable, therefore, to conduct a vibration test for at least the duration of the flight period of missiles.

As a consequence, the exaggeration factor which may be used in establishing the vibration amplitude for vibration tests of aircraft equipment can be minimized or eliminated for tests of missile equipment. There is a compensating influence, however, in the relative degree of reliability required. A guided missile is beyond the direct control of man during its flight time, and there is need for the maximum attainable reliability. Consequently, there would seem to be a need for a greater safety factor in both the design and testing of missile-borne equipment. Circumstances thus seem to call for vibration tests of greater-than-normal severity in any case, to compensate for the expected long flight-life of aircraft and to attain maximum reliability of missiles.

The primary record in the measurement of an environment is a trace of acceleration, velocity, or displacement as a function of time. If all of the vibration occurs at specific and defined frequencies, the spectrum of vibration is designated a discrete spectrum. On the other hand, the spectrum is designated a continuous spectrum if the vibration is distributed over a broad band of frequencies. If the spectrum is discrete, the amplitudes and frequencies may be obtained by inspection of oscillograms, or by use of a mechanical or electrical analyzer. In many instances, it is not possible to obtain a completely definitive plot of amplitude versus frequency, even though the spectrum is discrete, because the amplitude at any given frequency may have a different value at each cycle of vibration. If the spectrum is partially or entirely continuous, the possibility of defining the environment in terms of amplitude and frequency becomes much more remote.

It has been suggested that possibly the only valid simulation of an environment is one which reproduces explicitly the time history of the actual environment. Theoretically, this can be accomplished readily. It is done by recording the time history of acceleration on a magnetic tape and playing the signal back through an amplifier to the conventional electrodynamic vibration testing machine. Successful reproduction of the signal requires an amplifier whose response is flat over a very wide frequency band. Furthermore, the mechanical impedance of the load is a function of frequency and varies as the signal changes in frequency content.

As a consequence, the amplifier requires an additional degree of intelligence to compensate for this change in mechanical impedance. This intelligence has been provided in some instances by using a compensating network. The network is synthesized by first subjecting the equipment being tested to a scanning test in which the frequency is varied between lower and upper limits. The resulting amplitude of the testing machine may be interpreted in terms of mechanical impedance. The network which is synthesized on the basis of the measured impedance

is used with the amplifier to bring about necessary changes in amplification level.

There appears to be a valid reason to question the usefulness of this reproduction technique. It is possible to obtain a record of vibration experienced during flight only after the aircraft or missile has been built, flown, and instrumented. By that time, much of the equipment intended for use therein has been built, or at least the design has been crystalized to the extent that a quick change is not feasible. In view of the rapid technological obsolescence of modern weapons, it seems possible that the aircraft or missile would become obsolete before the effect of the reproduced environment is felt in equipment design. For this reason, it may be more appropriate to generalize than to reproduce an environment.

Effective generalization of an environment is impeded by a lack of understanding of its nature. A typical time history of acceleration as measured in a guided missile is shown in Figure 1. It is evident from inspection that this record includes many frequencies, but neither the amplitude of any frequency component nor the amplitude of the composite trace is obvious.

By plotting power as a function of the center frequency of the bandpass filter, the power spectrum is obtained. The power spectrum does not define exactly the severity of the environment, because the power as obtained in this manner is a function of the filter characteristics. To circumvent these uncertainties, the concept of power density has been introduced. Power density is defined as the power passed by an ideal filter having a passband width of one cycle per second; it is obtained by dividing the power by the bandwidth of the filter in cycles per second. The resultant quantity is designated the power spectral density, and the unit commonly used is g^2/cps.

An example of a curve of power spectral density derived from measurements in a missile is given in Figure 3. This plot provides information on the relative level of vibration throughout the frequency spectrum, but does not include information on acceleration amplitudes. A full description of the environment would include information on the distribution of acceleration amplitudes in the filtered trace represented in Figure 2, for each frequency setting of the filter.

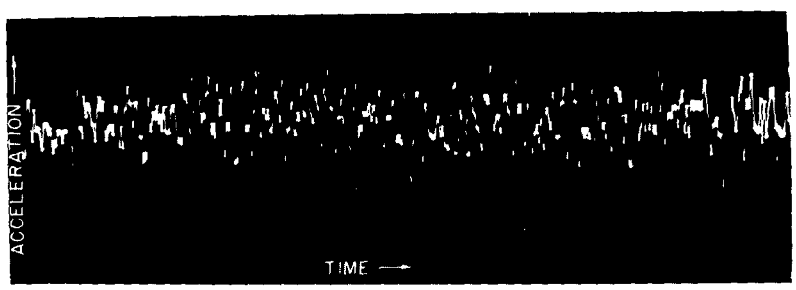


Figure 1 - Oscillogram of vibration in guided missile

If the record shown in Figure 1 is filtered by an ideal bandpass filter having a narrow passband width, the signal passed at any arbitrarily chosen frequency setting of the filter may have the appearance of Figure 2. The amplitude of this filtered trace varies with time. Its rms acceleration $\ddot{x}_m$ over the time interval from $t = 0$ to $t = T$ may be computed with the equation set forth above the trace. In an electric circuit, power is proportional to the square of the rms current. By analogy with this electrical circuit, the square of the rms acceleration obtained from Figure 2 has been designated the power of the vibration environment where the time interval from $t = 0$ to $t = T$ is sufficiently long to obtain a representative sample of the record.

Generalization of the environment, as by specifying power spectral density, requires that the nature of equipment failure be understood. If an environment is reproduced explicitly, it may suffice to determine only that failure of the equipment occurred or did not occur.

On the other hand, if an environment is simulated only generally, it is essential to define the environment in terms of parameters which are related to the potential to cause damage. The simulated environment must then reproduce these parameters. The widespread testing of military equipment is responsible for a great accumulation of vibration test results. Unfortunately, for present purposes at

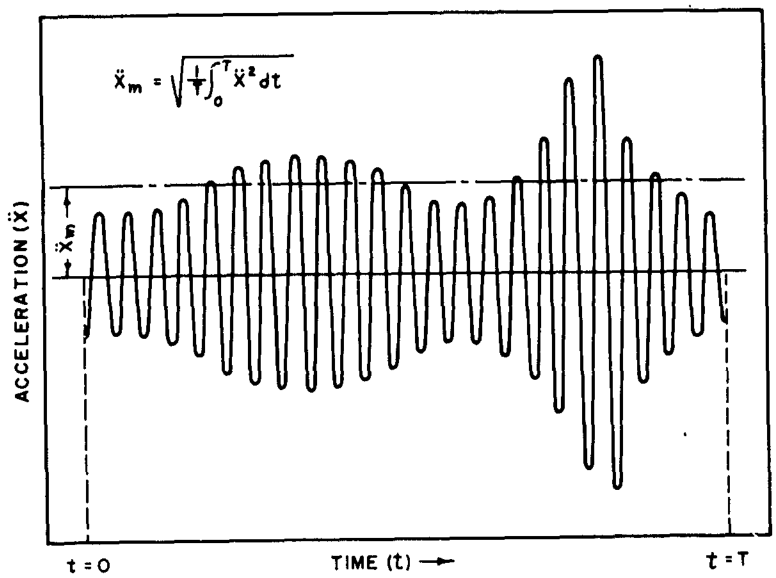


Figure 2 - Idealization of signal passed by filter

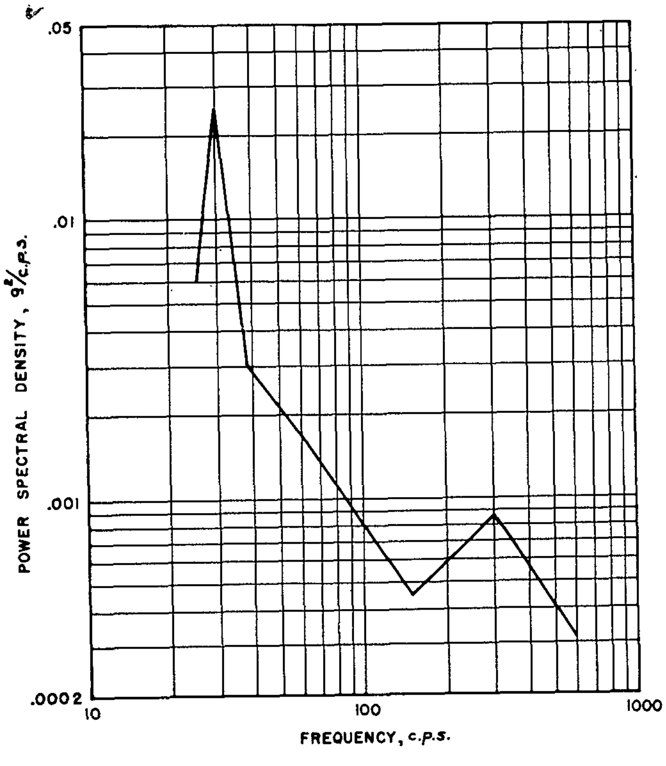


Figure 3 - Power spectral density in a guided missile

least, the preponderance of these tests have been conducted to determine conformance with established specifications. The test results are of a "go-no go" type, and little information has become available on the relation of damage to test conditions.

In the absence of tangible data defining the mechanism of equipment failure, an hypothesis may be assumed. This hypothesis visualizes the equipment as consisting of a complex array of structures having various properties and modes of failure.

For purposes of analysis, each structure is considered to be a damped single-degree-of-freedom system as illustrated in Figure 4. The

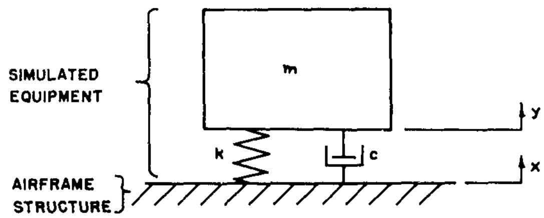


Figure 4 - Element of simulated equipment

diversity of structures comprising an equipment is simulated by considering the natural frequency of the above system to vary between wide limits. The likelihood of damage to the equipment is assumed to be directly proportional to the maximum stress in the spring of the system. This hypothesis is used to investigate the appropriateness of a testing procedure.

The simulation of an environment defined by a continuous spectrum can now be discussed with reference to Figure 4. From either the measured or the assumed environment, the power spectral density which describes the vibration of the airframe is defined. The maximum acceleration, or the distribution of the acceleration amplitudes, is not defined. The recent technical literature includes much discussion on the response of a damped, single-degree-of-freedom system to this environment, and it is possible to postulate physical laws from which information on the stress in the spring k may be computed:

1. A definite relation exists between the power spectral density of the airframe structural vibration x and the power spectral density of the simulated equipment y illustrated in Figure 4. The parameter which relates the power spectral densities is the square of the transmissibility ratio encountered in conventional vibration analysis.

2. The response of the simulated equipment is characterized by a tendency toward a normal or Gaussian distribution of instantaneous acceleration values. This tendency presumably exists independent of the distribution of acceleration amplitudes in the vibration of the airframe, particularly if the damping of the simulated equipment is small.

When the distribution of instantaneous values of acceleration is normal, the distribution of peak values of acceleration is described by Figure 5 wherein the ordinate is the probability of exceeding (or the probability of not exceeding) a peak value of acceleration, and the

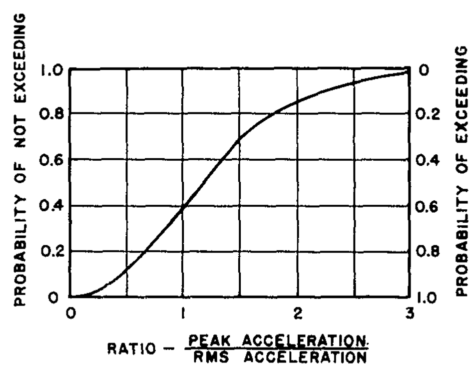


Figure 5 - Distribution of peak acceleration corresponding to normal distribution of instantaneous acceleration

abscissa is the ratio of peak acceleration to rms acceleration. The distribution curve in Figure 5 shows that 61 percent of the peak values of acceleration exceed the rms acceleration and 14 percent exceed two times the rms acceleration. It may be calculated that one peak in 268,000 exceeds five times the rms acceleration, and one in 5×10^{21} exceeds ten times the rms acceleration. Even though the value of the rms acceleration is of limited significance in mechanical design of structures, this distribution suggests certain limits, expressed as multiples of the rms acceleration, beyond which an acceleration peak is improbable statistically.

Before proceeding further with this discussion, we should remind ourselves that we are not attempting to reproduce an environment but rather to simulate it by devising a test which produces the same damage as the environmental condition. Referring to the curve of power spectral density illustrated in Figure 3, a possible simulation involves the reproduction of this density characteristic. Even though such a test does not necessarily reproduce the distribution of acceleration amplitudes which make up the power spectral density, it meets certain requirements of an acceptable test, because the response of the damped single-degree-of-freedom system which simulates the equipment, tends to adjust to a normal distribution independently of the distribution in the environment.

The power spectral density in Figure 3 represents only a single environment, however, and does not meet the condition of generality. It has been suggested that generality be achieved by assembling all applicable curves of power spectral density, and drawing an envelope through their peaks. It will be shown that such generalization drastically modifies the nature of the environment being simulated.

The response of an elastic system when subjected to vibration depends upon the damping of the system. For convenience of nomenclature, the damping is defined here by the parameter Q whose value is equal to the transmissibility at resonance when the system is subjected to a conventional vibration test at a single frequency. It is significant now to investigate the response of the damped single-degree-of-freedom system to two different types of excitation when both the natural frequency and the damping parameter of the system are considered to vary:

1. In a test at a single frequency in which the excitation frequency is varied slowly between the minimum and the maximum, a resonant condition exists when the excitation frequency becomes equal to a natural frequency of the system. The response of the system, expressed in terms of either the acceleration amplitude or the rms acceleration, is equal to Q times the corresponding quantity of the excitation. In other words, the response is directly proportional to the first power of the parameter Q.

2. When the excitation is characterized by a continuous spectrum, vibration at all frequencies presumably exists at all times with continuously varying amplitudes. As a consequence, any system whose natural frequency is within the range of the frequencies included within the spectrum experiences a continuous resonant condition. If the power spectral density is constant independent of frequency, the rms acceleration of the response is then directly proportional to the one-half power of the parameter Q.

These response data are summarized in Table 1. The relation between the power spectral density and the rms acceleration $\ddot{x}_m$ of the excitation are set forth in the second column for both discrete and continuous spectra. The third column gives the rms acceleration $\ddot{y}_m$ of the response. If Q is a constant, it seems evident by comparing the expressions in the first and second lines of the third column that the rms acceleration of the response can be equalized in the two instances by making the power spectral density in the continuous spectrum inversely proportional to the natural frequency f_n and directly proportional to the power spectral density of the discrete spectrum.

According to the hypothesis, an equipment consists of many systems, each with its particular value for Q. Because a different power of Q prevails during excitation by discrete and continuous spectra, it is not possible to simultaneously subject two systems of different Q to the same rms acceleration of response during excitation by both discrete and continuous spectra. This suggests that a continuous spectrum cannot be simulated by a discrete spectrum, if the rms acceleration of the response can be related directly to physical damage.

The relations given in the second line of Table 1 apply only to a continuous spectrum wherein the power spectral density is relatively flat as a function of frequency; i.e., wherein the curve of power spectral density does not include portions having large slopes. The curve of power spectral density illustrated in Figure 3, however, not only has slopes that are relatively great but also exhibits pronounced peaks. This indicates that the environment has characteristics of both discrete and continuous spectra.

It is possible to determine the rms acceleration of response of a damped single-degree-of-freedom system to this environment. Under these circumstances, the rms acceleration of response is proportional to the n^{th} power of Q where n has a value between one-half and one. If n is smaller than 0.75, the environment apparently can be better simulated by a test involving a continuous spectrum; if the value of n is greater than 0.75, the environment apparently can be better simulated by a test involving

TABLE 1
Summary of Responses of Damped Single-Degree-of-Freedom System

Spectrum	Excitation	Response
Discrete	$\ddot{x}_m = \sqrt{F(f)}$	$\ddot{y}_m = Q\sqrt{F(f)} = A_1 Q$
Continuous (Flat spectrum for $f_1<f<f_2$)	$\ddot{x}_m = \sqrt{\int_{f_1}^{f_2} F(f)df}$	$\ddot{y}_m = \sqrt{\frac{\pi}{2} f_n QF(f)} = A_2 Q^{1/2}$
Mixed	———	$\ddot{y}_m = A_3 Q^n$ where $0.5 < n < 1$

$F(f)$ = power spectral density, g^2/cps

$\ddot{x}_m, \ddot{y}_m$ = root mean square acceleration, g

f_n = natural frequency of responding system, cps

$Q = \frac{1}{2(C/C_c)}$ = damping parameter

a discrete spectrum, provided the rms acceleration is a significant criterion of damage.

In evaluating the relation between discrete and continuous spectra, it is prudent to keep the assumptions in mind. The maximum acceleration of a simple damped system was assumed as a criterion of damage, in the absence of tangible data on failures of equipment. The above analysis yields not the maximum acceleration, but rather the rms acceleration, together with a statistically defined distribution of acceleration peaks.

In a discrete spectrum involving steady-state vibration, the acceleration amplitude is equal to 1.41 times the rms acceleration and is directly proportional to the maximum stress experienced by the spring of the system. This acceleration amplitude is repeated at each cycle of vibration, and is of some significance in both design and test. In the continuous spectrum, the rms acceleration of response is essentially a mathematical value conveniently adapted for analysis. In attempting to infer its physical significance, it should be noted that, in the distribution assumed here, 61 percent of the acceleration peaks are greater than the rms acceleration and 37 percent are greater than 1.41 times the rms acceleration. In other words, 37 percent of the acceleration peaks in a vibration pattern whose peaks are constantly changing are greater than the acceleration amplitude of the system undergoing steady-state vibration with the same rms acceleration. It is thus difficult to determine whether the vibration experienced by the system with the constantly changing pattern is more or less severe than that experienced by the system undergoing steady-state vibration. It seems evident that considerably more research is needed to answer this question.

If the vibration environment in an aircraft or guided missile is to be simulated effectively, the environment must be defined by parameters which are significant physically and the laws governing failure of equipment must be established and confirmed. Insofar as piloted aircraft are concerned, a considerable mass of data exist, but data reduction means which have been employed leave something to be desired in understanding the true nature of the environment. Data which define the environments in guided missiles are much less plentiful than those defining the environments in piloted aircraft. There appears to be no preponderant evidence at this moment that such environments are either discrete or continuous.

On the basis of the preceding discussion, the following tentative conclusions may be drawn:

1. If the spectra defining the environments are continuous, there is a definite need to record typical and maximum values of power spectral density as a function of frequency, and to indicate whether the curves of power spectral density are predominantly flat or peaked.

2. An attempt to generalize a family of essentially peaked power density spectra by constructing an encompassing envelope

through the peaks results in the conversion of a discrete spectrum to a continuous spectrum. This generalization merges two types of spectra which have been shown in the preceding analysis of a hypothetical condition to produce essentially different results.

3. The statistical approach is inherently limited to defining the average value and the distribution of peak values. Many of the problems encountered in evaluating the strength of equipment subjected to vibration have not yet been solved for the condition of classical steady-state vibration at a constant amplitude. A useful understanding of the strength of equipment when subjected to vibration whose level constantly fluctuates does not exist at this time.

Finally, the simulation of vibration environments is currently in a very tentative state. Much more information is needed, not only on the nature of environments but on the strength of equipment when subjected to vibration, before it will be possible to state with some degree of certainty that a particular testing procedure is adequate to qualify equipment for use in any designated environment.

DISCUSSION

Mr. Rich, R.C.A.: The current vibration tests that we are designing right now specify a certain number of g's or displacement versus frequency over a specified frequency range. In Mr. Crede's paper, he mentioned power-versus-frequency testing. Would this be more advantageous than a g-versus-frequency testing since you always monitor the input at the table, and if you approach the resonant frequency of the load or equipment you can give a greater number of g input into the equipment than is going into the table.

Crede: This power density concept has come into the picture as a result of the continuous spectrum. Dr. Morrow refers to it as random vibration. The word "power," I am inclined to think, is a bit unfortunate because power is associated with watts in a lot of people's minds, and we are not talking about watts here at all. We are talking merely about a unit of the square of acceleration so that we should not make too big a distinction about the unit simply because we think of acceleration in one case and power in the other case.

It comes into the picture simply because the acceleration level is continuously fluctuating. About the only way you can reduce that to some simple numerical parameters is to measure the rms value of it. There have been some suggestions made that the proper unit to be used here is not g square over cycles per second, but rather g over the square root of cycles per second. In other words, take the square root of that power parameter on the assumption that most engineers are more used to thinking in terms of g than they are in terms of g square.

* * *

EVALUATING THE VALIDITY OF SHOCK SIMULATIONS

J. H. Armstrong, U. S. Naval Ordnance Laboratory

While it is not suggested that analytical solutions should replace testing in any but the most well defined examples on the simplest structures, in cases where knowledge of field conditions is incomplete or testing equipment inadequate, an exploration of simple dynamic and static relationships may be very useful in determining tests that will result in correct design.

The essence of valid shock simulation is that the tests reveal in the test specimen the effects that would result from the service condition being simulated. It is also necessary to refrain from producing effects that would not result from the service shock. When fed into the development process, failings in the first respect will show up in unreliability, while those of the second type will result in over-design.

Exact reproduction of a precisely-determined field acceleration-time curve by a machine whose output is insensitive to the mechanical impedance of the items it is testing is an obvious and ideal route to valid testing, but one which can rarely be traveled all the way. It is the purpose of this paper to explore the usefulness of some of the simplest dynamic and static relationships in pointing the way toward testing that can result in correct design and evaluation decisions in cases where the state of knowledge of field condition or specimen response is incomplete and where also, perhaps, the capabilities of testing equipment fall short of the mark in various respects.

This theoretical discussion, elementary as it is, is presented with no thought that analytical solutions should replace testing in any but the most well-defined cases on the simplest of structures. The tested mechanism alone is the true, quantitative judge of its own response, but from the study of stylized cases we may well hope to define simulation relationships that will permit the complex actual mechanisms to tell a true story. On the other hand, in most actual situations the profitable limits of the analytical approach extend far beyond the simple relationships herein discussed; perhaps this discourse may serve as a stage setting for some of the genuine advances and refinements in the analytical field to be reported upon later in this symposium.

DAMAGE SENSITIVITY

Excessive deflection of parts of a device under shock may be manifested in various ways: by microphonics in electronic gear, by chattering of a switch, by internal collision, or, in the common situation on which we will base most of this discussion, permanent deformation of parts from overstress causing subsequent loss of serviceability. Figure 1 shows a typical "damage sensitivity curve" for a structural element of a device or equipment. Adopting for the moment the bloodthirsty attitude that failure of a test consists of failure to damage the test specimen, it will be seen that in this plot of the velocity change of a single-shock pulse against the peak acceleration there are two ways in which the test item

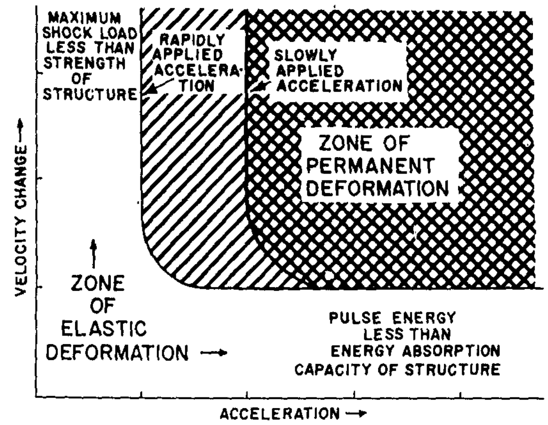


Figure 1 - Typical damage sensitivity curve

can escape permanent deformation.* If the maximum dynamic shock loading is less than the strength of the structure, or if the pulse energy going into the structure is less than its elastic energy storage capacity, there will be no damage.

EFFECTS OF RISE-TIME

As the plot indicates, rapidly-applied accelerations of half the slowly-applied value necessary to cause damage will also result in incipient permanent deformation, considering for the time being only cases where there is ample velocity change. For reasonably high-energy shocks, peak acceleration is the parameter usually considered basic. It is the one whose distribution is considered in studying, for example, the relationship between service stresses and component strengths in evaluating reliability. An uncertainty of two to one in this value is a little too much to ignore, even allowing for the fact that the usual variations in stresses and strengths are such as to reduce the value of extreme precision in the shock testing field.

The terms "rapidly" and "slowly" applied are not absolute, but are relative to the natural period of vibration of the structure subjected to them. To know in advance of firming up component design that a test is to be valid (and that is the very time when test procedures and facilities must be planned), therefore, the natural frequency of parts yet undreamed of must be established. Except perhaps in the tuning-fork industry, this is unusual. The acceleration value that is to be withstood, however, is usually established, at least to some degree, early in the game. By combining the equations for the bending stress in an element under inertial loading with those for natural frequency in the first mode,** as in the example of Figure 2, the frequency is found to be related directly to the acceleration by an expression of the form $f = K\,A/(c\varepsilon_y)$ where K is a constant typical of the class of structural element while c and ε_y are constants of the structural material. The frequency of any element, regardless of its size, is thus found to be directly proportional to the slowly-applied acceleration which will stress it to the yield point. In the example worked out for the cantilever beam of rectangular cross section, K has the value of 15.6 for acceleration in gravity units. As will be discussed later, for acceleration pulses having rise-times in the range to cause some amplification of the response of the element, the frequency will be increased by the response amplification factor η.

Figure 3 gives K values for six widely differing classes of structural element; note that K varies only between about 8 and 16. The value for the end-loaded cylinder is significant in that it is the lowest encountered, though from a practical standpoint this type of element is of importance only under extremely high acceleration such as in gun launching or plate impact. Now it can be stated that as soon as the minimum acceleration to be designed to is determined, for any given material, a floor has been set beneath which the natural frequency of any successful design of a self-supporting element cannot go.

EFFECTS OF CONCENTRATED MASSES

Most useful structural elements are not simple shapes free from concentrated masses,

*This paper is concerned with single-shock pulses; some of the principles may be applicable, with caution, to individual portions of more complex acceleration-time patterns. Because of this single-pulse assumption the effects of damping can be neglected.

**This discussion considers only the first mode, since this is almost invariably the cause of failure because of its larger deflections and stresses. Coupling of other modes and related structures will alter frequencies, but generally not to an extent affecting general relationships derived.

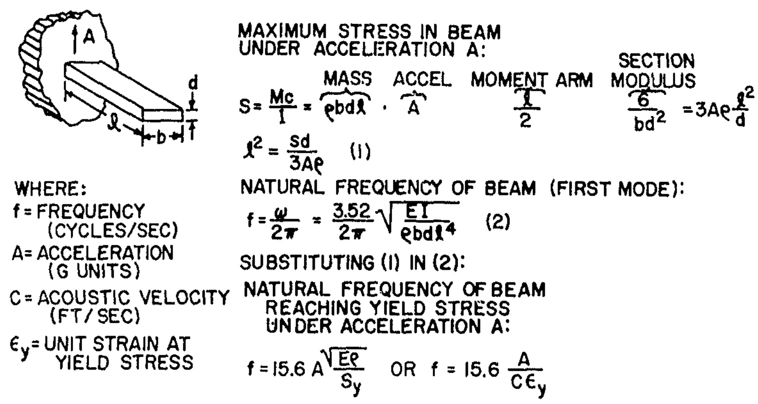


Figure 2 - Derivation of minimum natural frequency of a structural element resisting a given acceleration load

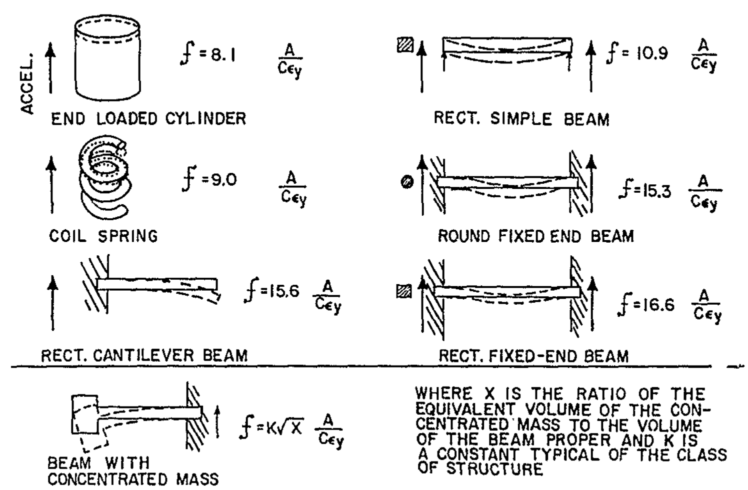


Figure 3 - Formulas for minimum natural frequencies of structural elements capable of resisting acceleration loads

so the case is also shown (at the bottom of Figure 3) for the cantilever beam with a lump at the free end. To reduce this to a system which can be handled in the general sense, the lump is represented as consisting of material of the same density as the structural part of the beam and having a volume X times that of the beam proper. In this and other concentrated-mass cases the frequency-acceleration relationship turns out to be similar to that for the beam alone* multiplied by the square root of the "concentration factor" X. Thus the "floor" under the frequency value holds true; the effect of any concentrated masses or other variations from the geometry of a simple structure is to raise the minimum frequency associated with a given acceleration resistance.**

*There is a small decrease in K, accounted for by the difference in stress distribution in the structural elements with the concentrated masses; in the cantilever beam case it amounts to about 2% when X is large.

**If X becomes very large (in the cantilever beam case, for example) the structure resembles a weight supported by a rivet and failure would obviously be by shear. Such cases, outside the scope of this discussion, are also covered by the frequency-strength relationship in that they will have much higher natural frequencies. During testing, they will "see" all shocks as slowly applied and will fail, with very little energy required, if shear strength is exceeded.

CRITERIA FOR SLOWLY AND RAPIDLY APPLIED SHOCKS

Figure 4 indicates how the relationship between frequency and strength establishes criteria for classifying shocks essentially as slowly or rapidly applied. The familiar response curve for a shock of indefinitely long duration indicates the amplification of the shock acceleration as a function of the ratio of the rise-time t_1* to the natural period T of the structural element involved. Most of the change in response occurs between the values of t_1/T of 0.25 and 0.80, where η decreases from about 1.90 to 1.25. Calling all shocks where t_1/T is greater than 0.80 slowly applied and considering η to be 1.10 will result in a maximum of about 15 percent error, while calling all cases rapidly applied where t_1/T is less than 0.25 and using 2.0 for η will be only about 5 percent off.

rule of thumb results by which the product of the peak acceleration (in g) and the rise time (in milliseconds) divided by the $c\varepsilon_y$ value for the material is less than 8 for rapidly applied shocks and greater than 80 for slowly applied ones. The $c\varepsilon_y$ term is a sort of "figure of merit" for materials with respect to shock resistance; there will be more to say about it later.

An example of a rapidly-applied shock would be a 50 g booster acceleration applied to a missile structure designed to be efficient in using its material to resist this loading. For a $c\varepsilon_y$ value of 100, typical of aluminum alloy, the rule-of-thumb value is 1.5 and a two-to-one dynamic load factor must definitely be considered for the critical elements if, for example, a static load equivalent is to be used in design or test.

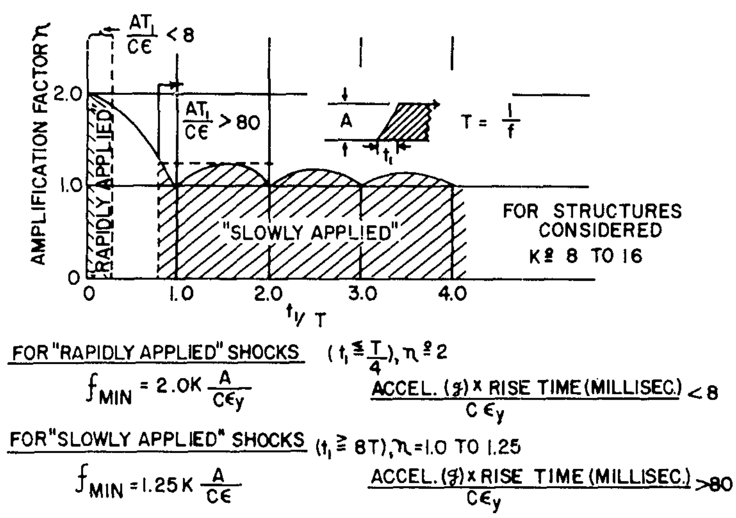


Figure 4 - Criteria for slowly and rapidly applied shocks

Plugging this into the frequency-acceleration formula and allowing for a variation in type of structure by assuming K values of 8 to 16, a

*The acceleration is assumed to rise linearly with time, as pictured; the differences with sinusoid or other types of rise are generally very similar if t_1 is selected as the time for the middle 80 percent or so of the increase to be accomplished.

If the rule-of-thumb value is 80 or above, the shock may be considered slowly applied with respect to the most "efficient" elements, those with the lowest possible natural frequency. As an example, a gun-launching shock on a projectile component reaching a 25,000 g level in one millisecond gives a rule-of-thumb value of about 100 with materials of the very highest $c\varepsilon_y$ value and is therefore essentially a slowly-applied shock which will not generally cause amplifications of more than 25 percent above static values.

EFFECTS ON OTHER THAN MINIMUM-FREQUENCY COMPONENTS

Any practical mechanism will also contain all sorts of structural elements, many of them designed on a basis of rigidity under operating loads, purely kinematic considerations, or other criteria unrelated to shock resistance—after all, a device which will withstand a million g but performs no usable function is of little value outside a museum. These heavier items will, for any particular level of shock resistance, have higher natural frequencies than the most efficient (shockwise) elements. They will therefore "see" any shock that is rapidly applied to the minimum-frequency components as, if anything, somewhat less than rapidly applied, as judged by the amplification with which they respond. If the shock on the minimum frequency components is slowly applied, all other elements will see the shock as even more nearly static in effect. The overall effect of this on the validity of test procedures can now be considered.

RELATIVE SERVICE AND TEST RESPONSE

From the simulation standpoint, the measure of test suitability in cases where for practical reasons it is difficult to determine or to simulate the service rise-times, is the relative response of all elements of the tested item to service and test pulses. Figure 5 illustrates two possible cases. In both, the service shock is of somewhat lower equivalent frequency (longer rise time) than the lowest-frequency elements in the test item, the rule-of-thumb value being about 60. For each possible element frequency—a range which extends only to the right from the minimum value determined by the basic relationship—the amplification factor for the service shock has been calculated.

In the upper example, the test shock is of still lower frequency; the "test shock" curve shows the response of the same elements to it. There is not much spread between the values, but in general there is some undertest, particularly in the range of the important lower-frequency elements.

In the second case in Figure 5, the test shock is applied about 50 percent faster than the service shock. Here, overtest predominates, reaching about 30 percent of the peak acceleration, though there are, at least theoretically, some small areas of undertest. Clearly, if the service shock frequency is this close to the minimum structural element frequency it is not possible to tolerate much variation in the test shock rise-time without introducing a 20 or 25 percent safety factor into the test amplitude and accepting corresponding overtest of some components. By considering only the portions of the curves to the right of 3.0 relative frequency ratio, which is the same as considering a case where the rule-of-thumb value is about 180, it will be seen that the situation will tolerate a good deal sloppier rise-time simulation without significant over or undertest. It is also evident

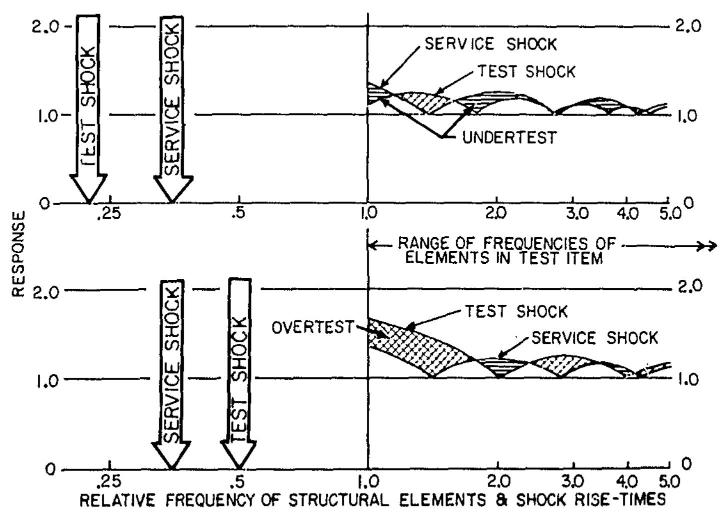


Figure 5 - Relative response to service and test shocks - I

that if the rule-of-thumb value on the basis of an under-estimate of rise time is well above 80, it is not necessary to measure the service rise-time closely.

In the first diagram of Figure 6 we consider cases where the service rise time is close to the minimum element frequency and the rule of thumb yields a still more intermediate, indeterminate value. Here it is clearly important that rise-time simulation be close. If it cannot be, the situation warrants closer study. Where the test shock is of considerably higher equivalent frequency, there is an area of overtest of considerable magnitude. Some slight improvement could apparently be made by reducing the test shock amplitude to bring balance at the minimum element frequency. There would still be much overtest, however, which, while all right from

> weakest parts of the design. In a sense, we are only required to have our simulation completely valid in the case of the marginal elements which may fail in service; the others can tolerate a certain amount of injustice.

The conclusion is that an incorrect test of this type (test shock frequency higher than service shock frequency) will not lead to unreliability and probably not to serious overdesign.

The second case in Figure 6 illustrates the converse problem, where the service shock is of more rapid rise-time than the test shock. Here the large area of undertest requires a considerable increase in test shock amplitude to steer clear of unreliability, with consequent

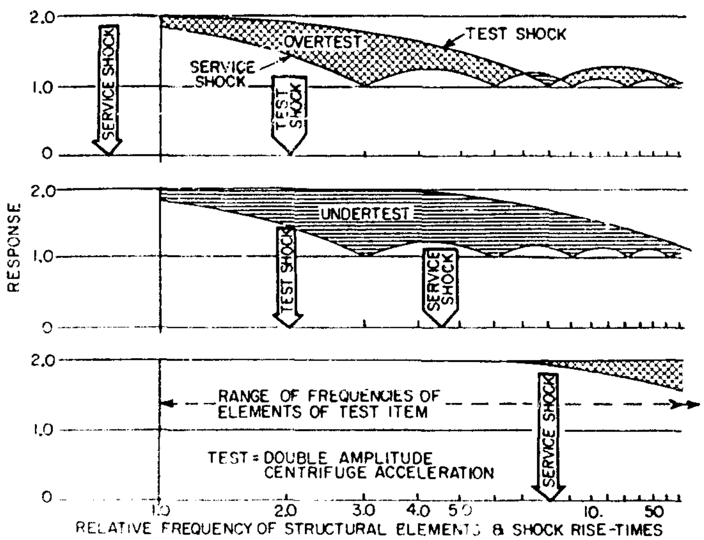


Figure 6 - Relative response to service and test shocks - II

a reliability standpoint, means a possibility of overdesign and reduced effectiveness. Most of the justification if such a procedure is used must come from reasoning of the following type:

> Many of the elements with frequencies higher than the minimum are still of "efficient" structural type; they are stiffer and stronger than the marginal components because they were designed to meet other criteria as well as shock resistance. Since they are by the same token stronger than the minimum, they will not fail under tests probing for the

severe overtest of the minimum-frequency elements. Since these are probably the most significant ones and should receive a valid test, the situation should be avoided.

If the service shock rise-time is very short in comparison to component natural periods (rule-of-thumb value is low) and cannot be matched with available test equipment, the best means of testing the structural strength of the item may be a double-amplitude centrifuge test. This results in a slight overtest, in the case of the highest-frequency components only, as shown in the bottom example of Figure 6.

OTHER SHOCK EFFECTS

The chief objection to the centrifuge test, of course, is that it will not dramatize the effects of internal collisions and transient out-of-phase deflections of parts which occur under shock loading and are often more important than those of maximum tension, compression or bending stresses. Such effects are the only ones considered in the above analyses.

To provide some idea of the likelihood of such problems and of the clearances which must be allowed between parts, the maximum deflection at yield of the lowest-frequency elements can be estimated, independently of the size of structure involved, using some of the same type of reasoning along with the relationship between natural frequency and static deflection. This maximum deflection is governed by a formula of the following form:*

$$\Delta = \frac{K'(c\varepsilon_y)^2}{A_{peak}}.$$

For components which need withstand only low accelerations but which are made of strong materials this number is surprisingly large (about ten inches for a 75 S-T column capable of withstanding 100 g, for example). This leads to the conclusion that internal collision is a far more likely source of failure than is structural bending, tension, and compression in structures which are designed to only low acceleration limits.

MINIMUM SHOCK DURATION

The preceding discussion has been predicated on service shocks of long duration, such as in high-velocity water-entry drag and missile boost cases, and on the availability of test methods providing ample pulse velocity change for full response. High-velocity change being synonymous with high energy, duration is often one of the limitations of simple shock testing machines such as drop testers which must be content with the one g provided by Nature. Another hurdle it would be desirable to avoid, if possible, is full-duration simulation of shock velocity changes exceeding Mach 1, a practical limit for pneumatic shock-testing devices. It will be of interest to see if some idea can be obtained of the test velocity change necessary to assure that some parts do not escape damage via the insufficient pulse-energy route (Figure 1).

A rapidly applied shock will have a lower peak amplitude in producing the same maximum structural deflection as a slowly applied pulse, and therefore less velocity change for a given duration, except as it may have a higher average-to-peak ratio. Often, too, the rise-time of a test shock is not independent of its duration. The interactions between these parameters in the response functions require that the situation be handled somewhat more carefully and that the effects of the dynamic response or amplification factor be taken fully into account.

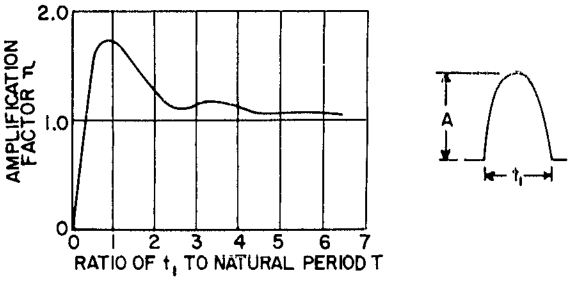
In Figure 7 the derivation of points on a damage sensitivity curve for a specific pulse shape (the half-sine in this case) is worked out, starting with the readily available response curve for the class of pulse.* The equation which results locates a curve that is independent of the absolute acceleration and velocity change. The velocity change at any point on the curve (ΔV) is dependent only on material properties, the class of structure (represented by the effect of the K value), and the pulse shape being considered (represented by the average-to-peak acceleration ratio and the t_1/T to η relationship of the response curve).

Figure 8 shows the resulting damage sensitivity curves for several pulse shapes. For all elements with linear force-deflection curves, the peak acceleration to cause yield is within the range of from 0.5 to 1.0 times the equivalent static acceleration for incipient yield. This variation is related to the rapidly- or slowly-applied pulse question previously considered and is properly handled by controlling (or at least rationalizing) the rise-time relationships between test and service shocks.

In terms of velocity change at the yield point, however, the curves for all pulse shapes, once the acceleration is past the static value for yield, lie along the same line where $\Delta V = V_0$, a value independent of the size of the specimen and of the acceleration which it must withstand. For the most efficient structure configurations where K is about 8, the value of V_0 is about $.65\, c\varepsilon_y$.

*Where K' is a constant typical of the type of structural element, roughly inversely proportional to the K values of Figure 3.

*The curve shown is for a concentrated-mass spring system; the equivalent curve for the distributed-mass systems more typical of this discussion is similar in shape but about 10 percent lower at the peak response.



1. THE RESPONSE OF A STRUCTURAL ELEMENT TO A HALF-SINE PULSE OF DURATION t_1 IS GIVEN BY THE CURVE.
2. THE MINIMUM NATURAL FREQUENCY OF AN ELEMENT STRESSED TO THE YIELD POINT BY A GIVEN STATIC ACCELERATION IS:

$$f = \frac{KA_{STATIC}}{C\epsilon_y}$$

$$T = \frac{1}{f} = \frac{C\epsilon_y}{KA_{STATIC}}$$

3. THE CORRESPONDING PEAK PULSE ACCELERATION IS:

$$A_{PEAK} = A_{YIELD\,(STATIC)} \cdot \frac{1}{\eta}$$

4. THE PULSE VELOCITY CHANGE IS:

$$\Delta V = \frac{A_{STATIC}}{\eta} \cdot \frac{A_{AVG.}}{A_{PEAK}} \cdot t_1$$

$$t_1 = \frac{t_1}{T} \cdot \frac{C\epsilon_y}{KA_{STATIC}}$$

$$\Delta V = \frac{A_{AVG.}}{A_{PEAK}} \cdot \frac{t_1}{T} \cdot \frac{C\epsilon_y}{K\eta} = .637 \frac{t_1}{T} \cdot \frac{C\epsilon_y}{K\eta}$$

Figure 7 - Derivation of damage sensitivity curve

"CHARACTERISTIC VELOCITIES" FOR STRUCTURAL MATERIALS

Since this $c\varepsilon_y$ term, the product of the acoustic velocity of the material and the unit strain at yield, appears recurrently in places of significance, it may be well at this time to digress a bit for discussion. Units of $c\varepsilon_y$ are feet per second; for want of a better term it will be dubbed the "characteristic velocity" for the material. Physically it represents the velocity to which the maximum elastic strain energy in a cube of the material could accelerate the mass of the same cube, without hysteresis loss.

Representative maximum values for several materials are given in Table 1. Materials likely to be used structurally in military equipment will have a characteristic velocity of about 40 ft per second or better; a good typical value for aluminum alloy structures is 100 ft per second, while such extreme combinations of strength, stiffness, and relative lightness such as music wire and glass-plastic laminates may reach values in excess of 200 ft per second.

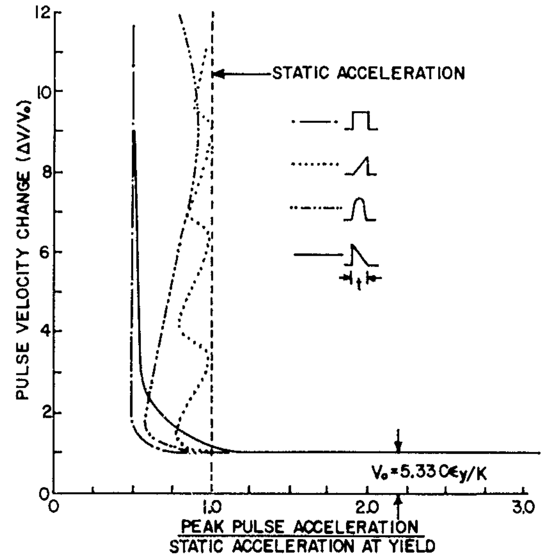


Figure 8 - Damage-sensitivity curves for various pulse shapes

TABLE 1
Maximum "Characteristic Velocities" of Structural Materials

Material	Yield Stress (psi)	Modulus of Elasticity (psi)	Density lb/cu in	Acoustic Velocity (ft/sec)	Characteristic Velocity ft/sec
Low-Alloy Steel	70,000	30×10^6	0.28	16,800	39
Tungsten	500,000	60×10^6	0.71	12,300	102
Aluminum Alloy 75S-T6	70,000	10×10^6	0.10	16,400	115
Methyl Methacrylate	17,000	0.5×10^6	0.043	5,600	190
Music Wire	400,000	30×10^6	0.28	16,800	224
Glass-Epoxy Laminate	87,000	4×10^6	0.067	12,600	277

CRITICAL VELOCITY CHANGE

The V_o value is not entirely suitable for use as an index of how much velocity change is needed in a shock test to insure valid simulation of a longer-duration service condition. In the first place, the damage sensitivity curve rises above the V_o value as the acceleration becomes less than the static yield value, that is in all cases where the shock is rapidly applied or applied at some intermediate rate where some degree of amplification is to be expected. The effect of this, as illustrated in Figure 9, in practical testing must be considered in terms of the equipment and procedures used.

In this example, the illustrated service shock pulse, assumed "square" (or at least of very fast rise-time and decay) and of 8.3 millisecond duration is simulated by three test shock patterns of similar shape but of shorter and shorter duration. As is usually the process, the empirical determination of the suitability of the device being tested is determined by applying successive shocks of increasing peak acceleration but constant duration. This is continued until something bends, breaks, or batters to an extent impairing its serviceability, or at least indicating incipient failure. The acceleration at which this occurs is compared with the service condition, with appropriate margins for the necessary degree of reliability.

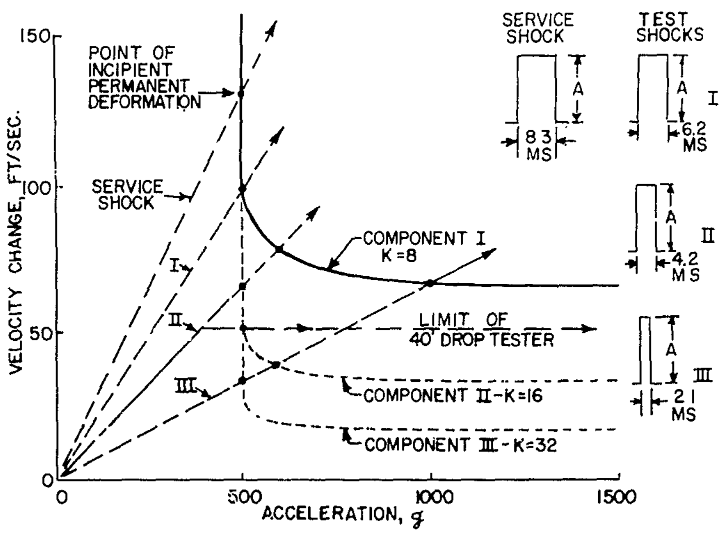


Figure 9 - Effects of test shock duration

The locus of test points making up a test program of this type appears on the diagram as a straight line emanating from the origin, with slope corresponding to the duration of the shock pulse. Since acceleration is being used as the independent variable, the measure of the validity of the procedure is the degree to which the test and service shocks result in incipient failure along the same vertical line of constant g. Assuming an aluminum alloy structure with $c\varepsilon_y$ of 100 ft per second, the service shock value for this duration is 500 g, resulting in a velocity change of 125 ft per second at yield.

Test Shock I, intersecting the highest damage sensitivity curve ($K = 8$) before it deviates from the vertical, produces incipient yield at the same 500 g as the test shock and is therefore valid. It does so with a velocity change of 100 ft per second, equivalent to a free drop of about 150 ft.

Test Shock II intersects the same curve at 600 g, 20 percent high. This in itself would not pose much of a problem—an adjustment in correlating field and laboratory values could be made. However, the actual test item will contain many other items with higher K values—structures of less efficient type or supporting concentrated masses. If they withstand the same shock, they must be of higher natural frequency.* Two examples (with K values of 16 and 32) are shown to illustrate the position of the sensitivity curves of such elements.

In addition, there will probably be some other elements made of less shock-resistant (lower $c\varepsilon_y$ value) material. Damage sensitivity curves for these are similarly displaced downward from those of their corresponding higher-strength counterparts of similar structure class. In a complex item, the area under the uppermost damage sensitivity curve may be visualized as being replete with curves representing all the many component structures.

The Test II shock will still intersect these lower curves at 500 g, so if the test acceleration were raised to 600 g to test the $K = 8$ component adequately, some of the other parts would be 100 g past failure. Though it reaches the V_0 limit at the service acceleration value, Test II is not adequate.

The shorter-duration Test III shock program will show a two-to-one disparity in effect among the various components. Since it would start deforming components at the 500 g value, it would indicate properly that something was wrong, but after these higher-frequency components had been redesigned, the lower frequency ones would remain in the device, to fail under the service condition. Still shorter pulses can show a greater than two-to-one spread.

*The K used here may be thought of as including the $\sqrt{X}$ factor (Figure 3) associated with components carrying concentrated masses.

LIMITED VELOCITY CHANGE EFFECTS

Returning to the Test Shock II curve, consider the likely course of testing if, for example, a drop test from a maximum height of 40 ft is being used. When the velocity change limit is reached, the only available course for raising the acceleration is to reduce the pulse duration, thus traveling along the horizontal path indicated. This will not cover the case of the low-K elements no matter how high the peak acceleration goes. Clearly, under these assumed conditions, the break point between adequate and inadequate velocity change is at 100 ft per second, the knee of the damage sensitivity curve.

This does not, of course, mean that lower velocity-change tests cannot be used for limited objectives, such as production quality control. The response of the actual item is the real criterion, and it may well be that the types of failure for which a screening test is needed are those such as brittle fractures of rigid components, shifting of bolted parts, etc., which do not involve the high energy typical of permanent deformations of strong parts of flexible configuration. However, it must be demonstrated by field tests or adequate-duration simulation that such is the case, and damage sensitivity analysis may help point the way to explanations for otherwise puzzling anomalies in response-versus-duration relationships.

COMPLEX RESPONSE CURVES

Considering again the damage sensitivity curves of Figure 8, it is evident that for other pulse shapes than the square one which was conveniently used for illustration the situation regarding minimum satisfactory velocity change is more ambiguous, particularly in the case of the curves that are multi-valued in ΔV. The relationships are properly studied by making comparisons between calculated response curves for the service and actual test pulses; these

usually indicate that a satisfactory relationship can be established if the "front face" of the test pulse (the rise-time) is kept close to that of the service shock, even though the total duration is modified in keeping with the capacity of the test gear. It may be noted that the fast rise-time pulses (⎍ and ⋀) are not multi-valued in ΔV.

PLASTIC DEFORMATION DESIRABILITY

It should be remembered that these damage sensitivity curves are for incipient permanent deformation. For a definite indication of failure, particularly if shock effects must be judged by go-no-go performance tests on a few samples, considerably more energy should be available. Doubling the pulse velocity change will provide energy enough for only about as much permanent set as the elastic motion prior to yield. The result is not too much of a safety feature.

On the other hand, the practical likelihood of K values less than about 12 is limited. It appears that a ceiling value for minimum acceptable test velocity change is about 400 ft per second, and that 200 ft per second should be adequate in most cases. This will insure (if the rise-time is about right) picking up any internal collision effects and (if the device will be altered in characteristics or rendered inoperative by small deformations) providing indication of damage. A few mechanisms may be so constructed as to accept some permanent deformation and remain operative; under careful analysis it may be possible to gain an idea of the total deformation under very long accelerations by studying the cumulative effect of multiple tests.

CONCLUSIONS

It is concluded that relationships developed between acceleration resistance and natural frequency may permit satisfactorily close evaluation of the validity of test procedures involving vaguely determined, incorrect, or poorly-controlled shock rise-times with respect to the types of failure (structural deformations) most sensitive to variations in this parameter.

For valid simulation of very long duration accelerations, pulses of 200 to 400 ft per second velocity change can be demonstrated to provide adequate energy to show up any areas of failure.

* * *

SOME SPECIAL CONSIDERATIONS IN SHOCK AND VIBRATION TESTING

Charles T. Morrow, The Ramo-Wooldridge Corporation

This paper discusses briefly a number of special considerations in shock and vibration testing, the philosophy of smooth specifications, the testing of components versus parts, force versus acceleration or amplitude excitation, and the single frequency equivalent.

SMOOTH SPECIFICATIONS

The typical vibration specification is a simple statement in which some quantity closely related to acceleration is kept constant over a wide frequency range, or at least is free of any abrupt fluctuations. A single-frequency sweep is kept constant at 4 g peak, or a random shake at 0.1 g^2/cps. Yet frequency analyses of service vibrations usually yield jagged spectra. Why, then, should our test specifications be smooth?

One immediate answer to this is that a test specification must above all permit a feasible test. Even if the service environments are known in detail, there is a practical limit to the permissible complexity of the test procedures and the manpower necessary to keep track of them and carry them out. This frequently leads to tests that in some specific instances may be unnecessarily severe, but it is easier to handle these through deviations, as necessary, than to complicate the specifications. On occasion it can also lead to testing that is in some sense inadequate.

A second answer to the question is that to verify reliability one must test to the extreme service condition, and what is a valley in the spectrum for one service vibration may be a peak in another. In short, one should test to an envelope of the service spectra insofar as possible.

For one reason or the other, most of the specifications we write or work to will be essentially smooth, no matter how well the service environment is known. There may be special cases, however, where the fluctuations in the spectra are sufficiently simple and the valleys are sufficiently reproducible to make a more detailed specification useful and feasible.

THE UTILITY OF SPECTRA OF FORCE

Vibration specifications are usually written in terms of acceleration, velocity, displacement, or their corresponding power spectra. Shock specifications are likewise written in terms of some function of acceleration. The accompanying forces have been neglected. What would be the virtues and disadvantages of monitoring the service vibration in terms of force by means of a strain gage at the mounting points of a component, and simulating on the vibration table in terms of force, again monitored by means of a strain gage at the mounting points?

If the objective is only to play back recordings of service vibrations through a complex wave system, it makes little basic difference which approach is taken so long as it is consistent. Equalization of the system to a flat response, either with the accelerometer or

strain gage, compensates for the differences between the vibration table and the structure to which the component is mounted in service, so long as the same type instrument was used in service to obtain the vibration recordings.

On the other hand, if the objective is to test to a smooth envelope of the service spectra, some questions are raised about the loading effect of the component on the structure supporting it in service, or on the vibration table, especially in the region of resonances in the component if much mass is resonating. When the component is mounted to something rigid and massive, loading is slight and acceleration is the better criterion. When the component is mounted to something light and flexible such as the skin of an airplane or missile, the loading effect tends to be large, and force may frequently be a more realistic criterion.

TESTING OF COMPONENT VS PARTS

In specifying tests for use during the design and development phase, at least, one would like to require an envelope of the service spectra, and one would like to do this for components, assemblies, subassemblies, and so on down to parts. But in practice it is usually not feasible to do this for more than one level of subdivision, and more difficult for parts than for components.

For example, with conventional chassis construction, it is frequently not feasible to prescribe a vibration test that a vacuum tube has a reasonable chance of passing, and that will guarantee reliability regardless of the chassis to which the tube is mounted. Of course, a partial test is useful in insuring that the part will be as good as possible, consistent with the current state of technology, but reliability is not independent of the way in which the part is used.

To pursue the example further, a chassis of conventional construction subject to vibration over a frequency range up to 500 or 1000 cps will have numerous resonances in the test range that may or may not coincide with resonances of the interior of the vacuum tubes. For simplicity of concept, this will be discussed in terms of single-frequency excitation, although the situation is similar, if perhaps less critical, for random excitation. At a resonant condition of the chassis, an amplification of 50 or more is not impossible before an applied vibration reaches the vacuum tube.

The relation between the chassis specification, the part specification, the fragility level of the vacuum tube, and the vibration transmitted by the chassis is illustrated in Figure 1. The vibration conditions specified for the chassis and part are shown as constant g single-frequency sweeps. The fragility level of the vacuum tube for a given frequency may be defined loosely as an applied vibration of such severity that, within some given short time, damage or malfunction will occur. All that is known about it from compliance to the part specification is that for every frequency it is at a greater severity than that of the part specification. At certain frequencies corresponding to resonances within the vacuum tube, the fragility level may be almost equal to the part specification severity. The vibration transmitted to the vacuum tube, when the chassis is tested to its specification, is shown as a jagged curve that exceeds the severity of the part specification at several frequencies.

If the fragility level and transmitted vibration are related as in Figure 1, the assembled chassis may pass its test. However, if on even a small fraction of the chassis the chassis resonances coincide closely with the vacuum tube resonances, these chassis may not pass test. The more severe the part specification by comparison with the chassis specification, the less probable this situation is. But if no control is exerted over the chassis design, it may not be feasible to raise the severity of the part specification high enough to prevent failure altogether. This would not be the most rapid way of attaining reliability in any case. If the chassis resonances were damped or raised above the critical range for the vacuum tube, the situation would again be less critical. Damping the part resonances would also be beneficial if feasible. In general, the effectiveness of damping is slightly less for excitation by random vibration and shock than by sinusoids.

It would be easier to design the chassis properly if the critical frequencies of the tubes and other parts as mounted were known, and preferably set by the manufacturer according to some appropriate convention, so that chassis resonances could be designed to avoid them.

THE SINGLE-FREQUENCY EQUIVALENT

The final topic that will be touched upon briefly in this paper is the problem of testing with a single frequency sweep when the vibration to be simulated is random. Anyone who has thought seriously about this problem will

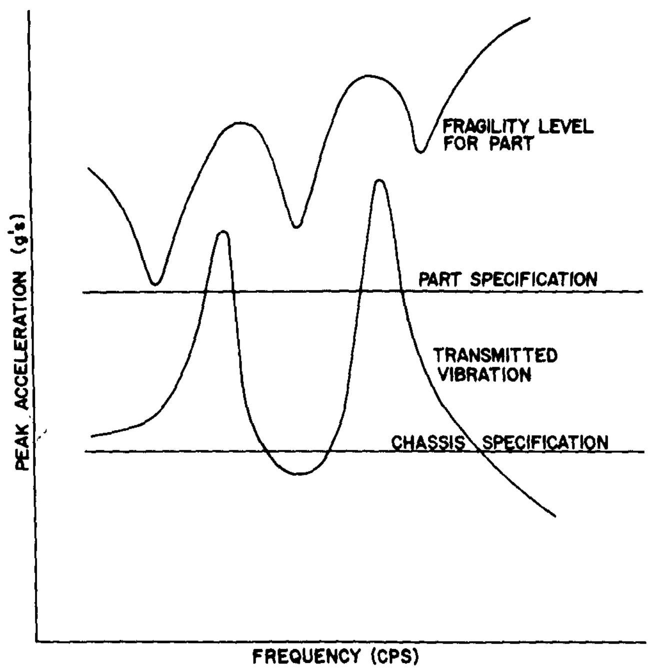


Figure 1 - Vibration excitations of a chassis and vacuum tube

agree that it is a messy one, and not easily solved to any satisfaction. The approach suggested here will not eliminate this aspect completely, but may help in some cases. We will assume that we are dealing with the test of a component, the envelope of whose service vibration spectra is known or estimated in terms of g^2/cps and contains no sustained sinusoids. A close simulation is desired.

Probably the most important factor here is that the peak applied sinusoid that will do the same damage to a resonant member as a given power spectral density in g^2/cps is dependent on the Q of the member, or in other words on the transmissibility of the part at resonance, or on how sharply it is tuned. If the number of g is set appropriately for the lowest Q in the component in a given frequency range, it will be too severe for higher Q's. If it is set according to the highest Q, it will be quite inadequate for others. If the objective is to attain or verify reliability, there is little choice but to set the single frequency specification according to an estimate of the lowest Q, or else if there are only one or two resonances of lower Q, test them specially. On the other hand, if this single frequency specification turns out to be hard to meet in the case of a given failure, it is desirable that the program not be delayed by this effort unless the effort is essential. It is desirable to measure the actual Q and frequency involved in the failure and readjust the test condition for the resonance accordingly.

Conclusion

The objective of an environmental test program is to maximize the rate at which reliability is obtained in the various portions of the schedule. Thus, the approach to environmental test must extend in scope beyond the mere solution of academic problems. Gambles must be taken from time to time beyond the limits of logic, and judgment must be used. A number of special considerations in shock and vibration testing have been discussed and, it is hoped, clarified helpfully even if not completely solved.

Whatever may be the shortcomings of environmental test as practiced, its influence has been predominantly beneficial. Whatever may be its ultimate limitations, it is certain to become of still greater benefit as its foundations are clarified and its techniques refined.

DISCUSSION

Markowitz, ERA, Inc.: I would like to ask Dr. Morrow if I understood him correctly. As I gathered, he was advocating that the vibration specifications be set at a quite high level to insure maximum reliability, and that in those cases where equipment gets into trouble, which possibly would be satisfactory in service conditions even though they fail to pass the vibration test, that waivers be set up. That is, set up a military specification and then take waivers to it. On what basis can you establish such waivers and who is to determine when such waivers are justified? How can we further qualify such a test to make it workable?

Morrow: Well, waivers to military specifications are nothing new, and there are various conditions in back of this. For example, you have something which fails in the military spec test but you may have evidence that when you get it out in the field you don't have failures on this particular part, so you ask for a waiver or a deviation. There are various techniques.

Now what interests me at the present time a little more than anything else is missiles, and here in a sense we don't work the military specifications to the same degree as in the other military areas. In other words, there is not one controlling military specification, and what we usually do is write one and include as much as might be helpful from some existing military specification. In the case of a missile, one way that a deviation might come about would be a comparison of the test condition with what happens on the telemeter record.

Now, of course, one can go too far. We can test a component with one shake, one play-back of a missile telemeter record, and say, "Hurray, we passed this one. It will pass all of them. There is no longer any need to pass the synthetic test." This is going a little bit too far because missiles aren't that reproducible.

But at least you can draw certain conclusions from the telemeter record. For example, if there is not much vibration in the critical frequency range wherein the given component is required to operate, it might be logical to consider deviation. Even if we have all the data that we would like to have about missile vibration, and we never do, we still wouldn't be able to take all of it into account in the specification. We just plain can't make the specification that complicated and we do have to have some safety valve for specifications.

Lusser, Redstone Arsenal: If I understood the speakers right, their papers concerned the principle of production environmental testing, in contrast with development testing—the latter being fully justified. Production environmental testing is supposed to screen out the items which are unreliable or cannot stand the condition. Those that pass go into the missile. But in fact, all the testing does is make them more susceptible to fatigue and subsequent failure in flight.

From the point of view of reliability, I feel that it is very much a matter of safety factors or safety margins achieved between stresses and strains. For instance, the safety factors in piloted aircraft are of the order of magnitude of 6, 7, 10, or 15. I wish to emphasize that we believe the only way to achieve the extreme degree of component reliability required in guided missiles is to apply very high safety margins on a statistical basis.

Morrow: In the first place, I don't think we were talking clearly about production testing. None of us actually did make a statement as to which kind of testing we were talking about. The problems of simulation are pertinent to both, although they have to be interpreted with a slightly different weight.

The thing which I had uppermost in my mind was what is frequently called developmental testing, and I would like to make a point here. Of course in setting up a missile activity we would like to make certain reliabilities maybe mandatory. But when you get down to anything much smaller than the whole missile you can't verify these things very well. You have to have certain substitute requirements with their appropriate exceptions that still can be demonstrated. Furthermore, as we go from an airframe to a component we find that we pass from an area where design is relatively easy and test difficult to an area where design is difficult but test is relatively easy.

Accordingly, the design techniques are much better developed for the airframe, whereas it is very common to throw things together, take a chance, try it out on the vibration table, and patch it up later in dealing with components. Up to a point this is frequently a saving of time because some of the things we throw together are so complicated that a complete dynamic analysis is impossible. On the other hand, I might add that it would be nice to know what to

do when the thing falls apart. Sometimes we don't. But at any rate, in terms of having something that gets things going and that can be demonstrable within the present state of the art, we have to depend primarily on tests at the present time so far as components are concerned.

In consideration of the number of samples that can be tested we have to put our factor of safety primarily into test conditions rather than into the margin with which the thing passes tests. Not that test to destruction beyond the specification requirements is not a good thing. But we do have to set up some standard of what is minimally acceptable. So this is a partial answer to your question, Dr. Lusser.

My factors of safety would be partly included in the test conditions. However, I hasten to add that I am very much in favor of production testing. I lived for several years with a missile program in which this was the primary type of testing that was carried out. Such testing is very good with a missile, and my experience indicates we are very much better off than if we had nothing at all.

We might say that a watch is something we wouldn't test this way, but a watch is a pretty rugged device, and frankly, I would test a watch with a production vibration test if I thought we would gain enough by it. It also happens that a watch is a very reliable device so there is no need to test the watch. But it is quite common in automobile manufacture, for example, to have a guarantee period in which we hope most of the failures take place and then to remedy them. We drive the thing around, try to get as many miles on within the guarantee period as we can, maybe so we get all the bugs out, and if we are lucky after that it ought to perform very well.

Likewise, we fly our missiles within the guarantee period before they have been shaken down, and production testing is a possible way of getting one a little bit further along toward the end of the guarantee period. It is desirable, of course, to use some moderation in this kind of test, not knock things apart, but if something is just hanging together we would like to know it.

Lusser: I would just like to point out in answer to Dr. Morrow a few examples to illustrate my point. The Bureau of Ordnance conducted a big investigation to find out if range running of new torpedoes did anything to improve their reliability, and as we shall hear in a paper tomorrow, I believe, similar tests were carried out with vacuum tubes. The results clearly showed that production environmental testing did not give any improvement in reliability.

* * *

PHILOSOPHY IN THE SHOCK TESTING AND DESIGN REQUIREMENTS OF SHIPBOARD EQUIPMENT

S. M. Blazek, Bureau of Ships

Questions are frequently asked about the meaning of the shock requirements of military specifications for shipboard equipment. The answers to these questions involve a discussion of the philosophy underlying the shock requirements, and a consideration of the methods of meeting the requirements by laboratory simulation tests or by calculation.

At past symposiums, papers have been presented on the characteristics and magnitudes of mechanical shock to which shipboard equipment could be subjected during time of war. These papers were supplemented with high-speed movies showing the types of motions and damages various items of equipment experienced during controlled underwater explosion tests.

During the discussions of the shipboard shock and its effects, a question on the meaning of the shock requirements in the military specifications was invariably posed. In an attempt to answer that question the following will be presented and discussed in this paper: (1) the philosophy underlying the shock requirements for shipboard equipment and (2) the methods of achieving these requirements by laboratory simulation tests or calculations. In addition, some general shock design considerations will be presented to aid the designers in obtaining shock resistant equipment.

Mechanical shock is an undesirable environmental condition encountered by the Navy ships in performance of their military function. Unfortunately, the cause or initiation of shock cannot be eliminated and, therefore, must be accepted as one of the "end conditions" in the design of the Navy ships. In brief, shock is a complex dynamic loading applied for a short time duration, milliseconds. It can be generated by:

1. Firing of the ship's own guns,
2. An aerial bomb explosion,
3. An underwater non-contact explosion.

The shock loading initiated by the underwater non-contact explosion, (3) above, is, in general, the shock considered in the design of shipboard equipment and discussed as shipboard shock at these symposiums. This loading is more severe than that initiated by (1) and (2).

The dynamic loading associated with a direct hit from an aerial bomb, torpedo, etc., is not considered as one of the conditions for design. The shock loading and associated fragmentation in the immediate localized vicinity of the direct hit and explosion is destructive, but the effects are negligible in the adjacent surrounding areas of the ship.

War damage reports and controlled underwater explosion tests have shown that shipboard equipment is more vulnerable to damage from shock loadings than the hulls of the ships. This means that the ships are more apt to be incapacitated by failure of equipment than failure of the hull for a non-contact underwater explosion. Therefore, it is necessary to improve the resistance of shipboard equipment to dynamic loading conditions—shock—in order

to increase the military effectiveness of the Navy ships.

The ideal goal which the Navy is attempting to attain in regard to shock resistant equipment is to have the equipment as shock resistant as the hull of the ship. That is, the vital shipboard equipment should not fail for any shock loading which does not rupture the hull of the ship.

It has been shown (War Damage Reports and controlled underwater explosion tests) that, in general, shipboard equipment which is not specifically designed for shock fails at much lesser shock loadings than the hull can withstand. Therefore, it is necessary to design shock resistance into the equipment. For this reason, the shock requirements have been incorporated into the "GENERAL SPECIFICATIONS FOR SHIPS OF THE UNITED STATES NAVY" and the individual purchase specifications for equipment.

The shock requirements as given in the specifications are:

1. Equipment weighing 4,500 lb or less is required to pass the shock test in accordance with the procedures of military specifications MIL-S-901 or MIL-T-17113 for general equipment and electronic equipment, respectively.
2. The subbases, feet, hold-down bolts and main structural members of the equipment weighing over 4,500 lb are required to be checked by standard static calculation methods to assure that the stresses do not exceed the yield strength of the material using the appropriate design factors given in Figure 1 for the three principal directions.

The shock tests are conducted on either the light-weight or medium-weight class HI (High Impact) shock test machines following the procedures outlined in the test specifications. The light-weight machine tests equipment weighing up to 250 lb and applies the shock loading in three directions. The medium-weight machine tests equipment in the weight range of 250 lb to 4,500 lb and applies the shock loading in only one direction.

The details of the testing procedure will not be discussed herein; instead, the purpose and meaning of the test will be discussed.

The high-impact shock testing machines are used to determine the shock resistance of equipment. Equipment that passes the shock machine test without damage, structural failure, or maloperation, is labelled shockproof and considered satisfactory for shipboard installation.

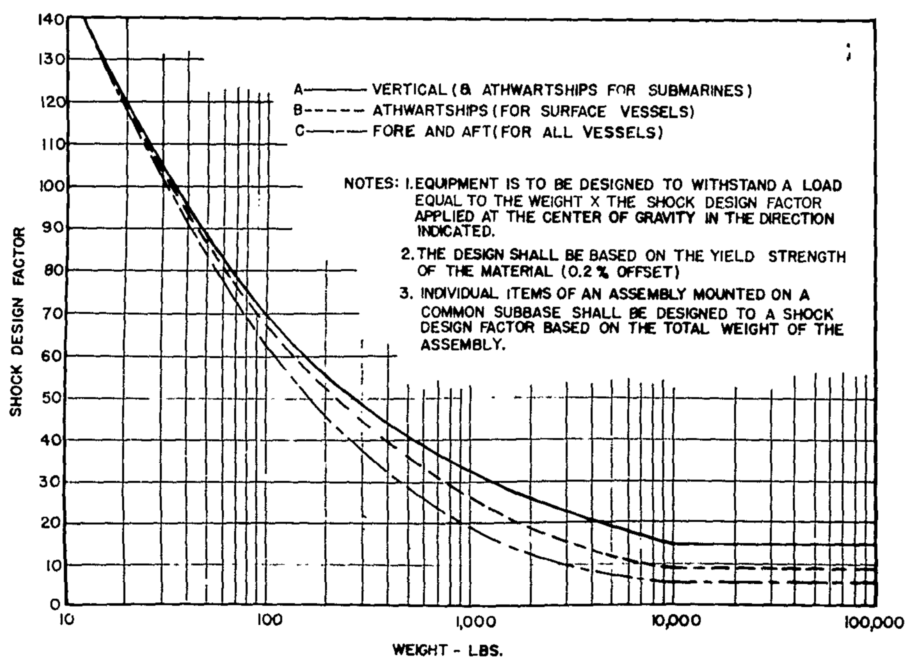


Figure 1 - High impact shock design data for hold-down bolts, feet, and main structural members of shipboard items of equipment

When built during the early part of World War II, these machines were considered to subject the equipment under test to shock motions whose nature and severity reasonably corresponded to the shock motions encountered under the severest conditions in naval service. The data obtained since then shows that the shock motions of the machines are not severe enough although the characteristics of the motion are, in general, realistic.

It has been found that some items of equipment, which successfully withstood the shock generated by the testing machines, withstood more severe shipboard shock conditions. This is not an unreasonable expectation since the testing procedure only requires that the equipment under test withstand a minimum level of shock loading and does not determine the maximum level of shock loading that the equipment can withstand. No doubt if a survey were made to determine the maximum level of shock severity that equipments passing the shock test could withstand, many items would withstand much more severe shock loadings. This is not to be interpreted to be a justification of why equipment should only be tested to the severity of the testing machine level and no more. It is only an explanation why some equipments withstand higher shock loadings in service than the shock machine severity.

Since the shock machines are expensive to build and serve to establish satisfactory standards for determining "shockproof" equipments, no changes to the machines are contemplated at this time. Instead, effort is being directed at the design of a machine capable of testing equipment weighing up to 20 tons. The paper by Mr. Gareau discusses the status of development of the heavy-weight machine.

The present machines operate on the same principle. The equipment under test is secured in a manner typical of its mounting aboard ship to an "anvil plate" which is struck by a rapidly moving hammer of considerable weight. The resulting motion of the anvil plate imparts to the mounted equipment a shock motion which the equipment should withstand without failure.

The equipment is never mounted directly to the anvil plate. Equipment is attached to the anvil table by means of a standard mounting adapter on the light-weight machine and by means of a standard mounting platform for the medium-weight machine. It is the intent of these standard mountings to approximate the actual rigidity or the most rigid mounting on which the equipment would normally be placed aboard ship. As an example, the 4A mounting adapter for the light-weight machine is used to simulate bulkhead mounting of equipment. The mounting flexibility alters the input shock loading to equipment and should be incorporated in the shock tests to simulate shipboard conditions.

The motion experienced by the anvil plate during the test embodies a suddenly acquired velocity which continues substantially without diminution until the plate has hit its arresting stops. Then it rebounds and gradually returns to its equilibrium position. The impact of the hammer against the anvil plate excites transient vibration of the anvil plate and mounting adapter. This vibration is superimposed upon the over-all motion of the anvil plate.

On both machines, it is required to test the equipment under its various operating conditions which, in general, are running and at standstill. The reason for this is to uncover any maloperations as well as to test the structural strength of the equipment.

The test blows are divided into three groupings with each succeeding group being more severe. This enables the designer or test engineer to make corrections that are required before severely damaging the equipment which could happen if only the severe blows were applied. It is not the intent of the shock test to be a repetitive or fatigue test.

Light-weight equipment is tested in three directions of equal severity because it is not known how the item may be oriented in the ship. The medium-weight shock testing machine is a unidirectional machine. This is not as serious a drawback as it may seem. In general, the orientation of the heavier items of equipment on board ship is known, and hence it is not necessary to test for the fore and aft direction which is a direction of minor loading.

For surface ships, the vertical direction is the most severe loading with the athwartship loading being only about one-half of the vertical loading. In general, if the equipment for surface ship installations is designed to satisfactorily pass the vertical shock test, it is satisfactory for the athwartship and fore and aft directions.

For submarine equipment, the severity of shock loading that can be attained for either the vertical or athwartship directions is nearly the same.

By virtue of the general design procedures employed in the shock design of equipment, the

equipment passing the vertical shock test, in general, will also be shock resistant in the athwartship direction. Some items of submarine equipment are mounted on angle brackets in order to apply the loading simultaneously in two directions when the shock resistance in the horizontal direction is critical.

Since no shock machines are available for testing items of equipment weighing over 4,500 lb, they are required to be stress checked. Static calculations are made using the shock design values obtained from Figure 1. In applying the shock design factors, only the hold-down bolts, feet, subbases and main structural members are analyzed for all three principal directions. For each principal direction, the static design factor is determined and then multiplied by the weight of the equipment to determine the force which is assumed to act at the center of gravity of the equipment in the direction considered. The stresses at the required places are then determined by applying standard calculation methods. These stresses are not to exceed the yield strength of the material(s).

For a few critical items in nuclear submarines, a dynamic analysis is required. The dynamic input specified in those cases is based on data obtained several years ago during an intense full-scale testing program. Unfortunately the dynamic analysis is complex and time consuming; however, it is worthwhile as more thorough analysis is made and the end result is more reliable than that obtained from the static design method.

When one or the other minimum shock requirement is met, namely, passing the shock test or satisfying the stress requirements by calculations, the equipment is labelled "shockproof." This means that the equipment has incorporated an arbitrary minimum level of shock resistance. But it does not imply that the equipment will withstand all of the shock loadings that can be encountered on board ship.

The only way to determine this is to use each type of ship as a shock machine subjecting it to the maximum shock loading it can withstand under controlled test conditions. Unfortunately, this is generally impossible.

The ships under test during the controlled underwater explosion trials are used as shock test machines for equipment and systems and for obtaining data on the character and magnitude of shock loadings.

From the instrumentation data obtained during those explosion tests, the following is being done:

1. A heavy weight shock testing machine is being designed,

2. Inputs to be used in a dynamic analysis are being determined,

3. The present static design factors are being reviewed and revisions considered.

As a result, the static factor curves for the hold-down bolts of submarine equipment will be increased. In all probability the general calculation requirements for submarine equipment will be revised in the near future after further investigations have been made on the effects to equipment in terms of cost, time, size, and weight.

Full-scale tests have indicated that the weakest component of shipboard equipment is the hold-down bolts. This certainly is about the easiest component of the equipment to strengthen.

Since it is virtually impossible to design a reasonable bolting area which will not stretch under shock loading, designers should be liberal and include more bolt area than that just required to meet the minimum shock requirements of the specifications by testing or calculations.

Failures occurring during shock loading may be in the form of a structural failure such as broken bolts, bent shafts, etc., or a maloperation such as tripped circuit breakers, open fuses, etc.

The modes of failure by machinery and equipment under shock can result in:

1. Relative motion between different parts of the same item. This may result in excessive strain at points where there are large bending moments causing either permanent distortion at the parts of the material, if ductile, or fracture of the parts, if material is brittle. Also, such motion may lead to failure of mechanisms, collision between separate components, crossing or opening of contacts which result in incorrect operation of the equipment.

2. Passage of stress waves. The passage of an intense stress wave with its associated strain through the material of any item may cause failure in its path if the stress wave is of sufficient intensity. In brittle materials, this would cause fracture

but in ductile materials, the yielding at any point in the path of the stress wave may result in distortion and hence cause maloperation.

The failures occurring to items of equipment during tests on either the shock machines or on ships during controlled underwater explosion tests, is the result of poor shock design. Design guides are not listed in the specifications because the purpose of the specifications is to state the requirements which the equipment must meet and not present the design details of achieving those requirements. Therefore, the designer's experience must be relied upon in designing shock resistance into the equipment. The following design guides are presented for the designer to follow keeping in mind the causes of failure mentioned above:

1. Hold-down bolts should be at least as strong as the main structural members, feet, and their means of attachment.

2. Split-ring lock washers should be provided with hold-down bolts.

3. Items should be secured positively and not by friction.

4. Open slotted holes in mounting feet should not be used.

5. Parts fitting into sockets or clips such as vacuum tubes, fuses, resistors, etc., should be self-clamping or be provided with auxiliary clamping devices to prevent dislodgment due to shock.

6. Where alignment must be maintained between two items of equipment, the items should be installed on a common rigid subbase.

7. If possible, center of gravity mounting should be used.

8. Brittle materials should not be used. Do not use cast iron. All materials should be capable of yielding appreciably without fracture.

9. Overhung components should be avoided; that is, avoid canterlevering masses.

10. Assembly of parts requiring a fixed relation should be positively secured to prevent serious displacement.

11. Pivot parts and linked mechanisms should be statically as well as dynamically balanced as far as possible.

12. For assemblies such as linkages attached to shafts, positive linking or position means such as pins or keys should be used. Pressed or shrunk fits alone usually are not sufficient.

13. All latching surfaces of latches must be well fitted. At times, such latching surfaces can be improved by the use of a negative rake angle so that slight additional force is required to open the latch.

14. Do not mount an item to two different main structural members that can move relative to each other.

15. Use generous-size fillets. Avoid abrupt changes in cross sections.

16. Stress concentrations should be carefully avoided. This may be caused by sharp corners, sudden changes in cross sections, sudden change in rate of change of cross sections, or sudden changes in material.

17. Adequate clearance should be allowed between fixed and moving parts and between all semi-rigid parts to prevent distortions under shock causing collisions, thus increasing the risk of failure or incorrect operation of the equipment.

18. Unit constructions should be used in mechanisms, i.e., components should be assembled as units and secured to common base plates or brackets. Parts of a mechanism should not be fixed to different parts of the case.

In summarizing, it should be remembered that items of equipment are designed basically to perform certain functions, then altered and improved upon to meet the environmental conditions. Shock machines are the best available yardsticks for accepting or rejecting items of equipment with regard to shock resistance. The above listed design guides, if followed, would greatly increase the shock strength of equipment.

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ENVIRONMENTAL VIBRATION TESTING FOR SHIPBOARD EQUIPMENT

Edward F. Noonan, Bureau of Ships

This paper reviews the requirements for vibration testing of shipboard equipment specified under Type I - Environmental Vibration, in MIL-STD-167(SHIPS). Information is given to provide a better understanding of the importance of environmental testing, and some of the reasoning behind the development of the specification is discussed.

INTRODUCTION

It is fundamental in this age of mechanized warfare that a piece of equipment be able to function properly when needed. This is equally true whether the equipment is a missile, a torpedo, an aircraft engine, or a gun director. It can be readily appreciated, therefore, that all military equipment must not only perform its function but also must be able to perform this function in the environment where it may be employed.

There are today a large number of engineers in the Defense Department, in Government, and commercial laboratories, as well as in our defense plants who are directly concerned with the development of specifications or the testing of equipment for compliance with environmental requirements. A large proportion of these engineers are primarily concerned with the environmental requirements for shock and vibration. To cover this field thoroughly, we have had a full program for each of the past twenty-two Shock and Vibration Symposia and will probably continue to do so for an indefinite period.

In this article, we shall briefly review the requirements for Vibration Testing of Shipboard Equipment as specified under Type I - Environmental Vibration, in MIL-STD-167(SHIPS) issued 20 December 1954. Some pertinent information will be given which may provide a better understanding of the importance of environmental vibration testing for shipboard equipment and point up some of the reasoning that went into the development of the specification.

In developing Type I - Environmental Vibration, as given in MIL-STD-167, a conscientious effort was made to devise a test procedure which would give us reason to believe that equipment capable of passing the specified tests would be capable of withstanding normal shipboard vibration. Very briefly then, we may say we are attempting to simulate shipboard conditions.

THE NATURE OF SHIP VIBRATION

To explain properly what might be considered the normal level of shipboard vibration, it would be best to start with a few words on the nature of the shipboard vibration we are considering. Fundamentally, we are considering the hull as a vibrating platform to which we must attach our equipment. The ship as a whole is equivalent to a free-free beam which has vertical, athwartship, and torsional natural frequencies. A ship is an extremely flexible structure.

The first frequency is normally encountered between 1 and 2 cycles per second with the frequency of the higher modes occurring at approximately equal multiples of the fundamental. Excitation of this structure may be occasioned by wave slap, unbalance in the propulsion system

or hydrodynamic forces acting on the hull at a frequency equal to the number of propeller blades times the shaft rpm.

The propeller blade frequency is generally the predominant one and determines the important range of frequencies to be considered. Upon examination, we find this range to extend from zero to 33 cycles per second, a frequency which may be found on some of our late submarines. Table 1 gives the maximum amplitudes expected for the ranges of frequencies shown.

TABLE 1
Amplitudes of Vibration

Frequency Range (cps)	Amplitude (plus or minus in)
0 - 5	up to 0.10
5 - 15	.030
16 - 25	.020
26 - 33	.010

These amplitudes were chosen as being representative of the upper values to be anticipated in the main structural elements of various ships in the fleet. Vigness and Hardy of NRL presented a compilation of frequency and amplitude data for typical naval surface ships which demonstrates the reasonableness of the above figures. These data were published in the Shock and Vibration Bulletin No. 21 of November 1953.

Although research on excitation of shipboard vibration is continuously supported by BUSHIPS and considerable guidance is given by the H-8 (Vibration Excited by Propeller Hydrodynamics) and H-6 (Hydro-Structure Vibration) research panels of the SNAME, we cannot hope to eliminate vibration aboard ship any more than we can hope to eliminate roll and pitch. We can, however, hope to reduce the levels of vibration anticipated. The U.S. Navy Training Film on "Shipboard Vibrations" is recommended for a more complete explanation of this subject. The film is in four reels and is identified by MN-9180-A to D.

THE ENVIRONMENTAL VIBRATION TESTS

Considerable discussion was held between representatives of the Navy Laboratories, Naval Shipyards, and Bureau of Ships to take full advantage of all experience on this subject in developing a specification which would serve our purpose with a minimum of alteration to the existing test procedures. Basically, MIL-T-17113(Ships) was found to be reasonably satisfactory for this purpose and, in addition, had been successfully used for a number of years in determining the suitability of many pieces of electronic equipment. An attempt was made, therefore, to take full advantage of the past experience accumulated by the laboratories and bring the requirements of this specification up to date.

There are three tests called for by MIL-STD-167(Ships) under Type I - Environmental Vibration: The Exploratory Vibration Test, The Variable Frequency Test, and the Endurance Test. In all cases, the equipment undergoing test is mounted on the vibration table in the manner in which it is intended to be installed aboard ship. If it is furnished with resilient mountings, it will be installed on the table with these mountings attached. The equipment should, if possible, be energized to perform its normal function. Each test will be conducted separately in each of the three principal directions of vibration unless this requirement is modified as noted below.

The Exploratory Test is intended to indicate the presence of resonant conditions in the equipment itself or in its mounting system. The amplitudes are purposely reduced to avoid unnecessarily damaging the equipment during this screening phase of the test program. The rate of change of frequency during this test is sufficiently low to permit careful observation of the equipment for selection of resonant frequencies. The primary purpose of this test is to determine the frequency or frequencies at which the equipment is most likely to suffer damage.

The variable frequency test is run at the estimated service amplitudes. The equipment is vibrated for five minutes at each integral frequency throughout the complete frequency range. This test is intended to reveal the presence of any serious resonance in a fully enclosed equipment or section of a larger piece of equipment which cannot be visually observed. Generally, deficiencies resulting from this test will be noted by malfunctioning of the equipment.

The endurance test is a two-hour run conducted at the resonant frequency or frequencies chosen by the test engineer as being the most likely to cause failure.

The amplitudes at which the variable frequency and endurance tests are run are given in

Table I of MIL-STD-167(Ships) and corresponds with Table 1 given above, for frequencies above 5 cps. The specification does not call for testing below 5 cps since very few items may be expected to have resonances below 5 cps and to extend the test down to zero cycles per second would present an unwarranted complication. Recent missile experience has indicated, however, that greater attention should be given to this low-frequency range.

It will be noted that the test amplitudes specified above 15 cps have been reduced from those called for by the original MIL-T-17113 (Ships). Although the frequency range has been increased because of our new vessels, the actual velocity of the test table does not exceed approximately 3 in/sec. The corresponding maximum acceleration transmitted is approximately 1.2 g.

Although the "full treatment" is recommended for equipment intended for use on all types of naval vessels, the specification does provide for a limitation on the upper range of frequencies to be tested for when particular items of equipment are intended for limited application. Thus, equipment intended for surface vessels only need not be tested beyond 25 cps. In like manner, equipment which may be intended solely for use on vessels having a maximum propeller blade frequency of 15 cps or less, need not be tested beyond 15 cycles.

To use this MIL-STD to maximum advantage, it is important for the engineer who is preparing the contract specification for the equipment to become familiar with the vibration specification, the intended use of the equipment, and previous experience obtained on similar items. At the discretion of the Project Engineer, it may be possible to omit certain parts of the test procedure and still comply with the intent of the specification. For example, it is often possible to omit the variable frequency test or to use it in place of the Exploratory Vibration Test if previous experience on similar equipment indicates this course is desirable. As another example, if an item to be tested happens to be asymmetrical in two planes, it may be possible to waive the tests in one plane.

Unlike the requirements for the shock tests, no particular vibration machines are called for in MIL-STD-167(Ships). Any machine capable of meeting the test conditions specified, is acceptable. The machine most widely used is the 500-lb, positive-drive vibration machine designed by Western Electric Company. Unfortunately, there are too few heavy weight vibration testing machines; machines capable of handling equipment of the order of 5,000 lb or better. NEL and NRL are equipped with heavy weight reaction type vibration tables. Some jury-rigs employing electrodynamic shakers have also been used on occasion.

SERVICE EXPERIENCE

Of the various environmental conditions found aboard ship, those of shock and vibration are among the more serious. At the same time, they are probably the most difficult to design against. The main reasons for this are that the parameters which must be controlled in any design are either too elusive or possess contradictory requirements.

Attempts to design delicate equipment intended for shipboard use frequently result in the application of shock absorbing devices at the points of attachment to the vessel. The apparent reasoning behind the selections made in many cases is the preconceived notion that mountings are required to attenuate the shock loading imparted to the equipment by the specified BUSHIPS shock tests. It is normally assumed that such tests are by far the more damaging to the equipment. As a matter of fact, it is easily appreciated how the average designer can ignore the importance of the vibration tests when he witnesses a shock test on a piece of his equipment.

The truth of the matter is, however, that the incidence of failure during vibration tests has been at least as great as that occurring during shock tests. NRL Report 4179, "Damages Resulting from Laboratory Vibration and High Impact Shock Tests" by Kenneth E. Woodward, summarizes the results of shock and vibration tests conducted on some 270 individual equipments. This report clearly indicates the need for designers to consider carefully the importance of the specified vibration requirements.

Failure to pass the environmental vibration test usually means the equipment malfunctioned or sustained some damage during the prescribed tests. Aboard ship deficiencies in equipments manifest themselves in much the same manner. Structural resonances in items of equipment may precipitate major casualties in the unit or result in its malfunction. Some particular gun directors for example have shown such characteristics. Resonances of components or items of equipment installed on mountings intended for shock absorption frequently result in fatigue

failures of parts or inefficient operation of the item or the system in which the item is employed. For example, the exposure of computer consoles to prolonged vibration at the resonant frequencies of their shock mountings can readily result in damage to the equipment in the form of loss of accuracy, too frequent replacement of components, or actual fatigue failure of parts.

Up to this point, the implication is that all deficiencies occurring in equipment aboard ship are the result of improperly designed equipment or a poor choice of mountings. This, of course, is far from the truth. Much difficulty is encountered due to improperly installed equipment or resonances in the supporting structure.

Structural sections, when supporting a relatively heavy piece of equipment, may represent a simple mass-spring system which has up to six degrees of freedom. Continuous operation at any such resonance could result in malfunction or damage to the equipment. Fortunately, however, modifications to eliminate such deficiencies in the supporting structure are usually amenable to correction while deficiencies within the equipment or the mounting system furnished are not so readily eliminated. Thus, it is important that the equipment, as furnished, be capable of withstanding the normal level of vibration encountered aboard ship, assuming the equipment is installed on an adequate foundation.

At present the Naval Research Laboratory is engaged in the preparation of a guide for the design of shock and vibration resistant electronic equipment. This publication will lean heavily on the experiences gained during past shock and vibration testing and should prove of significant value to manufacturers of such equipment. It is primarily for their benefit that the publication is being prepared. This document should be completed very shortly, providing no serious delays are occasioned by higher priority work.

CONCLUSIONS AND RECOMMENDATIONS

It is important that all equipment, structures or assemblies normally installed or carried aboard ship be capable of withstanding the environmental vibration conditions normally found aboard the ship. This is equally true whether the item is furnished by BUSHIPS as part of the vessel's basic equipment, whether it is fire-control or missile-handling system furnished by BUORD, or whether it is some particular type of weapon stored aboard ship until it is needed. Size or weight should not deter designers from considering this important aspect of service requirements. Even though it may not be possible to submit some equipment to the testing procedure outlined in MIL-STD-167, it is nonetheless imperative that adequate design considerations or testing of subassemblies be undertaken to insure the equipment not only remains intact but also performs its function satisfactorily when installed aboard ship.

One of the most frequent questions we encounter is, "What types of equipment should be vibration tested?" To this I would generalize the following categories:

1. All vital ship-control equipment,
2. All fire-control equipment,
3. All communication equipment,
4. All other electronic equipment, meters, gages, etc.

Ordinarily it is not necessary to vibration test heavy machinery items such as motors, generators, pumps, turbines, engines, etc. These items are normally sufficiently rugged to satisfactorily withstand shipboard vibration.

A few basic recommendations are given which should be of value to the specification writer, manufacturer, and design engineers in obtaining a satisfactory shipboard installation:

1. All equipment and structural assemblies should be designed to withstand the amplitudes and frequencies expected aboard ship.
2. All equipment which might be expected to be vibration sensitive in the frequency range of 0 to 33 cps should be vibration tested if it falls in one of the categories given above.
3. On important items of equipment vibration testing should not be waived because of a shortage of test facilities in the higher weight range. Unless the requirements for the heavy-weight test equipment are stipulated, the lack of facilities can be expected to continue.
4. Before resilient mountings are employed, care should be taken to insure they are required.
5. If mountings are required, care should be taken in their choice and application to

insure a vibration problem is not being created in eliminating a shock problem. For further information on this subject see DTMB report No. 880, "A Guide for the Selection and Application of Resilient Mountings to Shipboard Equipment," by Francis F. Vane.

6. Equipment normally carried aboard ship should be designed against its environmental vibration just as well as installed equipment.

7. When possible, vibration sensitive equipment should be installed in the center half of the vessel where the amplitudes are normally lower than observed at the bow and fantail.

8. Equipment should be installed on rigid structural members, whether or not resilient mountings are used.

9. Wider use should be made of vibration generators during construction to determine the presence of resonances which may be excited by propeller blade frequencies.

10. A free exchange of information between shipyard personnel is encouraged to obtain the best and most economical solution to shipboard vibration problems.

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DISCUSSION

Vigness, NRL: My question is, what is the harmonic content of the waveform of the vibrations measured on board ship? I think some of the later papers at this meeting will show that there may be quite a little harmonic content in vibration machines. I was wondering whether we have a corresponding amount of harmonics in the ship vibrations.

Noonan: On board ship we may encounter the fundamental frequencies of the hull which are occasioned by wave slap; a frequency which corresponds to shaft rpm for the first order frequency of the shaft; and the propeller blade frequency which is usually four or five times the shaft frequency.

As a matter of record, however, we have found out that in most cases the vibration encountered is predominantly the blade frequency. When occasionally we get a shaft frequency superimposed on this, it is generally associated with some derangement in the propulsion system such as an unbalance or hydrodynamic unbalance in the propeller. This is what we might call a deficiency, which we can track down and eliminate in most cases.

Vigness: Could you say how sinusoidal the waveforms were?

Noonan: They're very well shaped sinusoidal forms.

SIMULATING AND CONTROLLING RESPONSE OF A FUZE PACKAGE

Raymond W. Warren, Diamond Ordnance Fuze Laboratories

Two independent approaches which are being followed to obtain information for designing reliable guided missile fuzes, are discussed. One approach determines the vibration environment of the fuze in flight and attempts to simulate it in the laboratory. The other approach is to construct the fuze to function reliably under the full output of a vibration machine while being swept through the available frequency range.

Vibration studies are being conducted at the Diamond Ordnance Fuze Laboratories to obtain information for designing guided-missile fuzes that will operate reliably in flight. Two independent approaches are being followed. The first approach is to determine the vibration conditions in flight, and to simulate these conditions in the laboratory. The second approach is to design and construct the fuze so that it resists vibration. The fuze is then tested to determine the conditions under which it fails.

While it is desirable to simulate the vibration condition of a fuze in flight, this first approach has not been successful to date. However, as a secondary objective, production of an "equivalent damaging effect" would be satisfactory.

For these studies there are needed:

1. Reliable flight data on vibration amplitude, frequency, and phase in three principal planes.

2. A complex-wave vibrator for reproducing the flight-vibration data in the laboratory with the same amplitude, frequency, and phase in three planes.

There are numerous conditions which contribute errors to the "in-flight" vibration data of the fuze. The nose of most missiles moves in an elliptical path when driven by vibration. The electronic package of the fuze is usually located in the nose of the missile. The package frequently has a rocking mode when the mounts cannot be located properly or when the mounting structure is flexible. The problem is to measure the resultant complex acceleration in three planes, with an accelerometer which is sensitive in one plane but is also affected by accelerations in other planes. It is very difficult, if not impossible, to describe the acceleration of the fuze with the one or two telemeter channels that can usually be used for vibration information in a flight.

The systems generally used for obtaining a record of in-flight vibration over a wide frequency band have been investigated. The systems consist of a barium titanate accelerometer, a high-impedance cathode follower, a subcarrier oscillator, and a telemeter on the missile, as well as a receiver and tape recorder on the ground. It appears that the minimum errors in such systems are:

Amplitude	±8%,
Frequency	±12%,
Phase	±5% under 10 kc.

The maximum error can be very large, as any microphonics present generally appear on the record as an increase in amplitude and frequency. Work is under way to reduce the microphonics in the system in order to increase the accuracy of the vibration measurements.

While it does not appear that exact flight-vibration data can be produced on the complex-wave vibrators, they do have the following advantages:

1. A test can be programmed and automatically run on the vibrator,

2. The same test can be reproduced whenever desired,

3. The equipment is not subject to random high g at resonances.

Carefully controlled conditions are necessary to produce a sine wave on the object being vibrated. With an appreciable load on the table the waveform at best only vaguely resembles a sine wave, and a few inches above the table the side motion is frequently as great as the vertical motion. When vibrating a complex structure, it has been practically impossible to produce a sine wave at a tube clip.

Most vibration testing is done under room conditions, whereas in flight the temperature, pressure, and humidity are always varying. This can have an effect on the operation of electronic components during vibration. To study these effects an environmental chamber into which the table of a 3,500-lb-force electrodynamic vibrator is placed has recently been completed at DOFL. The frequency range is 10 to 2,000 cycles per second. A simulated flight can be programmed on this equipment with characteristics such as:

Temperature	-100 to +225°F,
Humidity	20 to 95%,
Altitude	Room pressure to 125,000 ft.

Calibration of this equipment is essentially complete and work has started on determining the effect of flight conditions on electronic circuits while being vibrated.

The second approach, of constructing a fuze which will operate reliably under the most severe vibration test which can be imposed by available equipment, has been more successful than duplicating flight vibration conditions in the laboratory; and while it undoubtedly results in overdesign, the equipment works in flight, which is most important.

In general there are two types of malfunction due to vibration in fuzes:

1. Physical breakage,
2. Microphonics which cause premature operation or failure to operate.

Physical breakage due to vibration is not difficult to control. Microphonics are much more difficult to cope with. As the major portion of the microphonic effects appear to originate in the tubes, an accelerometer closely simulating the size, weight, and shape of the subminiature tube was developed. This quickly showed that with typical sheet-metal and terminal-board construction and the usual tube clips, transmissibilities of 10 and 20 are not uncommon at the tube. Maximum transmissibilities of 5 have been considered satisfactory.

Several possible solutions to the preceding problems are evident from the graph shown in Figure 1.

1. Isolation mounting,

2. Damping,

3. Natural frequency at least double the forcing frequency.

There are many combinations and variations of these three approaches. However, considering the space limitations on a guided missile and the wide range of frequencies encountered, it appeared that either high natural frequency or damping promised the quickest success, so parallel work was started on each.

A program for improving the vibration performance of existing amplifiers was started. An i-f amplifier which was hermetically sealed in a sheet-metal can was vibrated while empty and then the can was filled with silicone oil. The silicone-oil-filled can, when vibrated, resulted in a reduction in microphonics. The results are shown in Figure 2.

The action of the oil on the tubes of the amplifier during vibration, and determination of whether the reduction of microphonics is due to the acceleration and deceleration of the mass or viscous damping has come in for considerable discussion. These problems have not been completely resolved. Tests indicate that use of the silicone oil is more effective at low frequency.

It is difficult to prevent oil from leaking out of a fuze. There are places in a fuze where oil cannot be permitted, in addition the oil also caused a frequency shift which, however, could be compensated for.

The next approach to applying damping was to fill the same amplifier with hollow ceramic balls to obtain coulomb damping. The balls

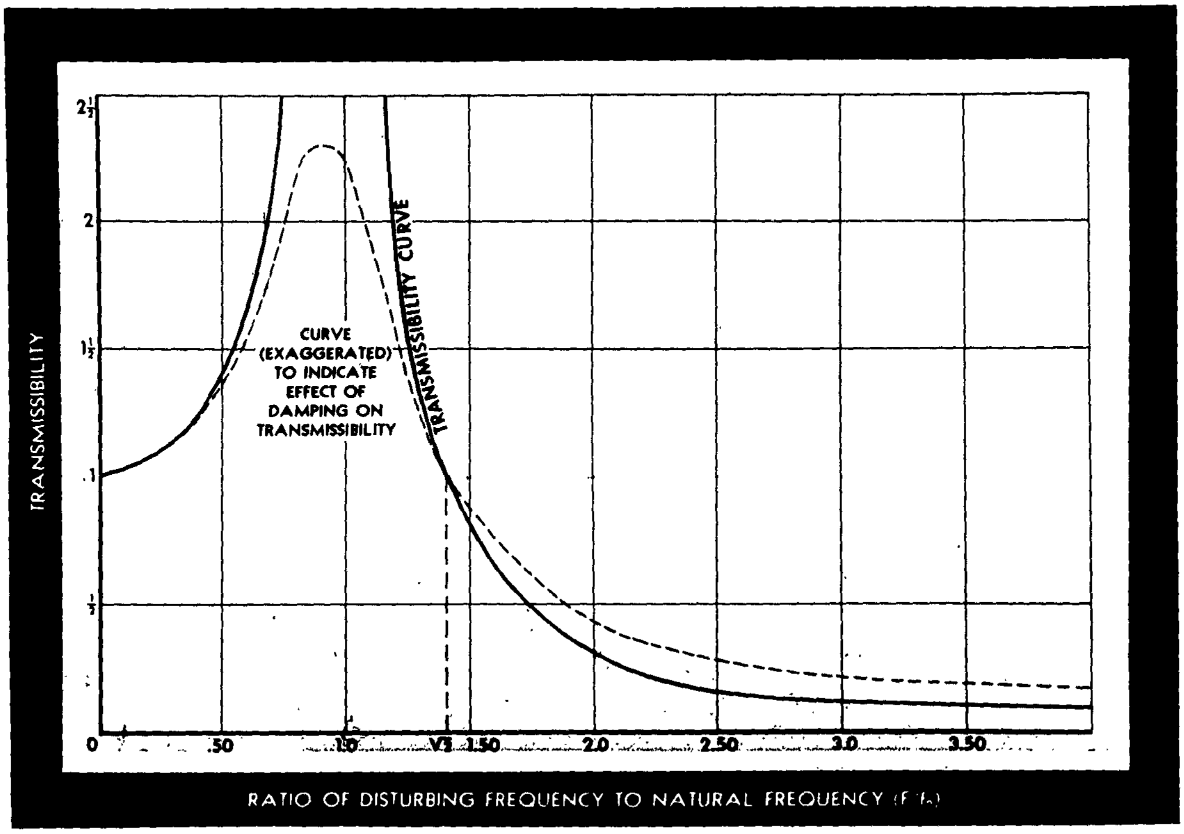

Figure 1 - Transmissibility vs. frequency

were of random size but all would pass through a 30-mesh screen. Their weight was 29 lb per cu ft. The results during vibration are shown on Figure 3. Unfortunately some of the clearances were only twice the diameter of the balls, so the damping effect was limited.

Next a complete fuze was filled with ceramic balls, with the results shown on Figure 4.

The ceramic balls did not cause a noticeable shift in frequency and caused no sealing difficulties. The damping effect of the balls is more velocity-sensitive than that of oil, being less effective at high frequency.

The approach used in designing equipment to have high natural frequency is to employ aluminum castings with cored holes for components. The components are secured in the holes with a rubber-base adhesive or attached with screws. The passive components in the circuit, which do not emit appreciable heat, such as coils, capacitors, and resistors, are grouped in modules and rigidly potted in epoxy resin (Figure 5). This eliminates the conventional terminal boards which frequently gave difficulty because of resonances. It also permits packaging in a considerably smaller space, resulting in no increase in weight even though the unit is potted.

Considerable hand wiring is eliminated by using printed wiring. Dip soldering to connect components in the modules, and mechanized methods of assembling the modules, can be used.

The electronic modules are cemented into holes in the aluminum frame, with a rubber-base cement. They can easily be removed and replaced through use of a few drops of benzine, gasoline, or methyl ethyl ketone. All connections are accessible for inspection, or are potted in plastic. The aluminum frame with the holes for the modules and tubes can easily be cast in production, eliminating considerable machining. These components and an assembly are shown in Figure 6.

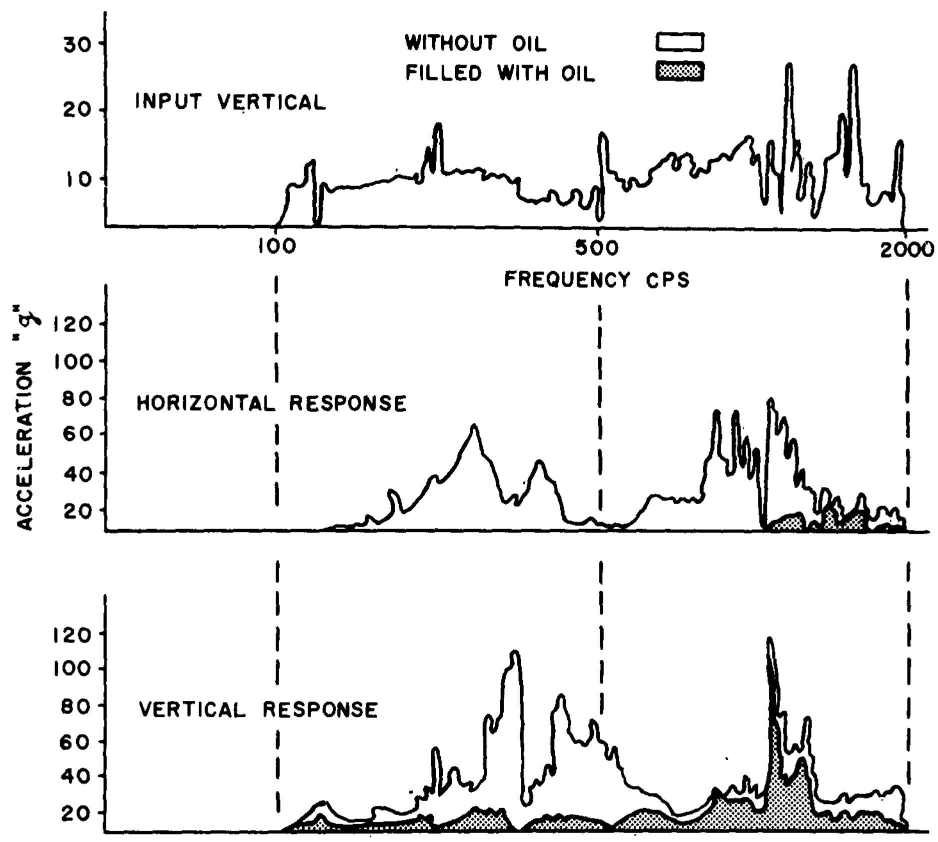

Figure 2 - Amplifier vibrated with and without oil filler

The subminiature tubes are held rigidly in aluminum mesh. The preformed mesh is first placed in a hole in the structure. A force of approximately 2 lb is necessary to insert the tube into the mesh. The response of the tube to vibration when held in this manner, is one-to-one within the limitations of measurement. Heat conduction through the aluminum mesh to the aluminum structure is excellent.

Working with structures that have a high natural frequency and a transmissibility approaching one, it soon became apparent that an accelerometer was of limited value in evaluating the structure. Also, if the transmissibility is low, then it is the microphonics or noise in the circuit which becomes of interest.

The same broad-band amplifier circuit was used for evaluating all rigid structures. The tube types, tuned transformer, and circuitry were selected because of a history under vibration conditions. The gain of the amplifier is 53 db.

The aluminum frame was secured directly to the table of the 650-lb-force vibration machine and vibrated from 10 to 2,800 cycles per second at 5, 10, 20 g and the full output of the machine.

The amplifier on the vibration table was fed with an r-f signal generator. The output was put through a vacuum tube voltmeter and recorded on a direct reading recorder.

The microphonics generated by the amplifier while being vibrated at various frequencies and amplitudes is approximately equal to the microphonics generated by the rigidly held tubes alone. The microphonics was negligible as far as fuze operation was concerned at full output of the vibration machine, which at many frequencies was in excess of 20 g.

In an attempt to determine the capabilities of the electronic modules, they were mounted in a beam with a natural frequency of 1,200 cps, approximating the natural frequency of the

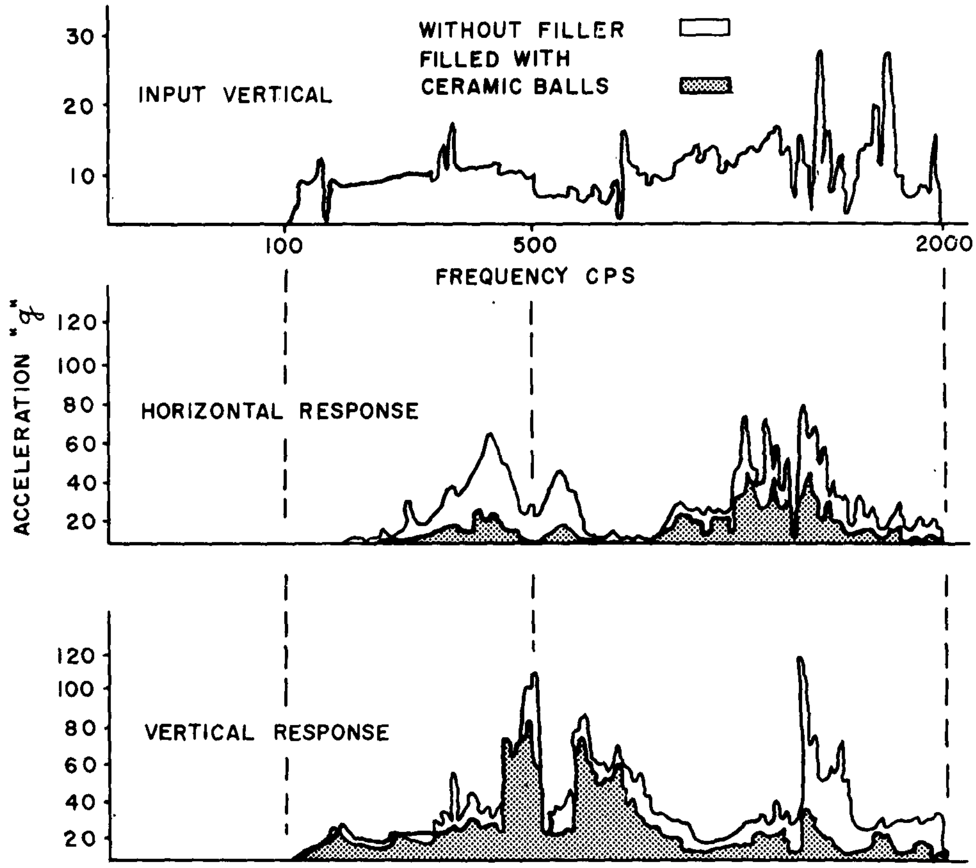

Figure 3 - Amplifier vibrated with and without ceramic ball filler

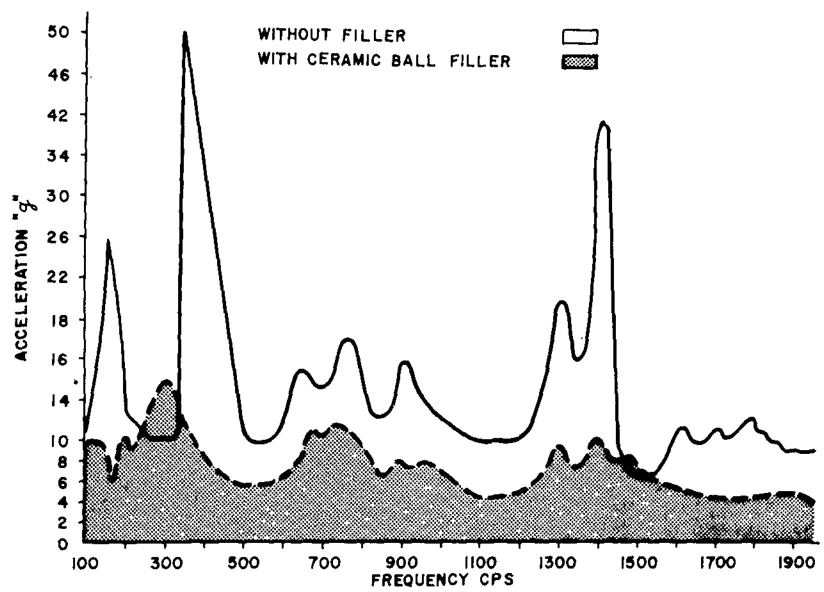

Figure 4 - Fuze vibrated with and without ceramic ball filler

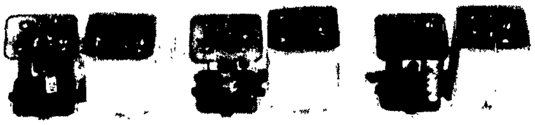

Figure 5 - Electronic modules

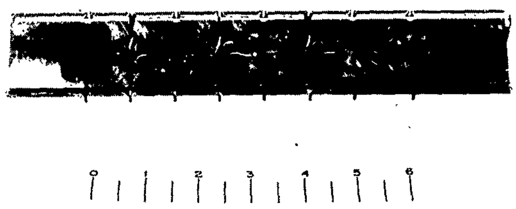

Figure 6 - Modules in frame

tubes. The beam was vibrated at 86 g, the maximum that could be obtained with the system. The microphonics generated within the system at 86 g was excessive and the trace went off scale. However, after vibration ceased, the unit operated satisfactorily. After continued vibration at 86 g the circuit operated, but generated more microphonics at lower g levels than it had previously.

The electrical characteristics of the unit were the same after 10 drops at 60 g and 10 drops at 100 g.

Figures 7 and 8 illustrate a simulated "fuze structure" designed to have a high natural frequency.

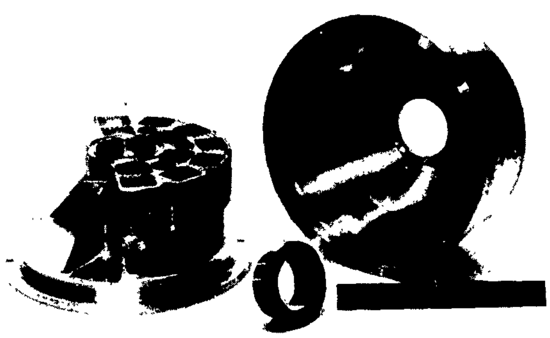

Figure 7 - Chassis for modular electronic circuits

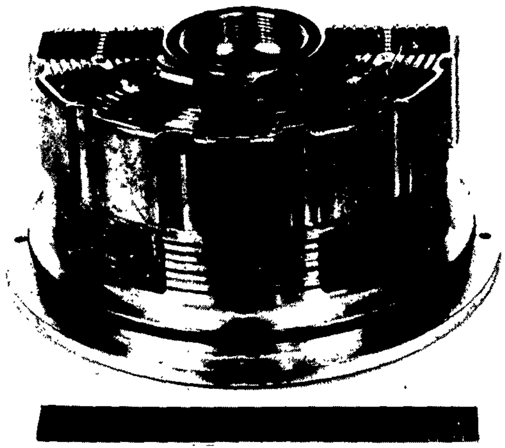

Figure 8 - Chassis with modular electronic circuit in place

The most microphonics usually are generated in the electron tubes, and the tube in the first stage of the amplifier is the most critical. With this interest in holding tubes, several methods of securing tubes in castings have evolved.

The vibration transmissibility of all these subminiature tube holders (Figure 9) is very close to one. The transmission of heat to the block is satisfactory for all the tube holders.

A high-natural-frequency structure is very satisfactory for fuze circuits during vibration. However, in the case of a very weak signal and a high-gain amplifier, even when the transmissibility is one, the microphonics may be excessive. An example of this is the high-impedance cathode follower commonly used between the accelerometer and the telemeter to obtain broad-band-vibration information in flight.

When vibration-resistant tubes or transistors are used, the microphonics due to vibration are reduced though not eliminated. Other items in the system, such as some types of transformers and capacitors, also contribute microphonics.

Experiments are now starting on a low-natural-frequency highly damped package for the components which produce the most microphonics due to vibration. This will probably include the first stage of an amplifier. The approach appears especially promising where it is necessary to achieve minimum microphonics.

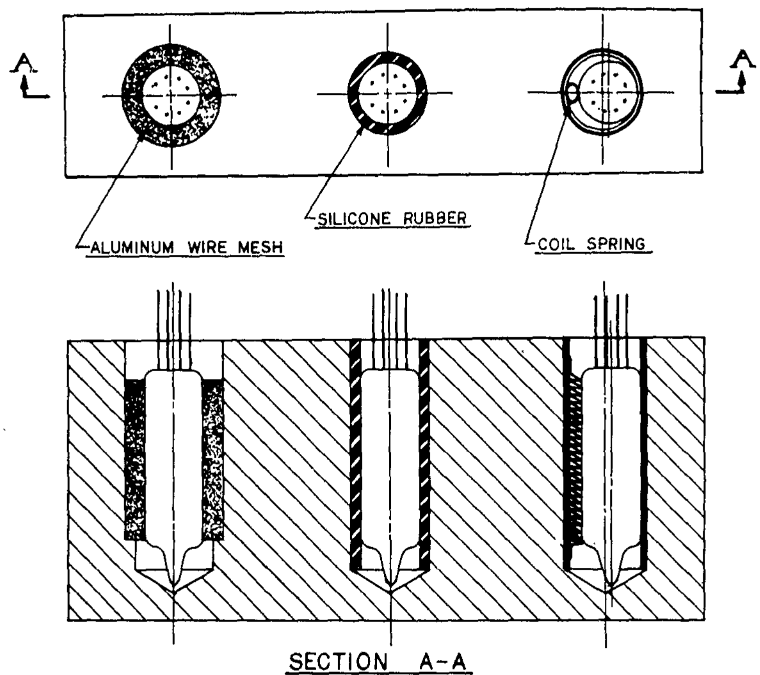

Figure 9 - Subminiature tube holders

DISCUSSION

A. Fine, NOL, Corona: I have a comment. We have recently issued a report, Technical Memorandum 62-110 and its supplement. You will find information on the TERRIER fuze in it, and we have reason to believe the information is correct because we have supporting information from captive land line, captive telemeter, and in flight.

In addition, we were fortunate in one test. The Diamond Ordnance Fuze Laboratories wanted a test run to determine the effect of a smoke puff charge on the fuze. At the time this was run, we disengaged the piston arm from the air spring, but we still had all accelerometers installed in the missile. The effective force came from the forward end and we got the transmission of the shock throughout the entire missile.

In addition to getting the information for Diamond Ordnance Fuze Laboratories, we also got confirming information on the transmission of the shock through the TERRIER structure and the reliability of our air spring system. This information on the fuze is available and can be obtained by writing to the Naval Ordnance Laboratories, Corona.

Warren: Thank you. The question of reliability, though, runs quite a bit deeper than that. You'd have to start with an accelerometer and carry it right through and analyze the system, and we believe that while we are getting the errors I listed in the laboratory, in flight they are probably considerably higher.

C. B. Cunningham, NRL: This seems to be an "I hate tubes week." I wonder if you contemplated the use of transistors and what luck you have had with them.

Warren: Yes, we have experimented with transistors and we have had very good results, but we have not eliminated the microphonics completely. Our noise levels with transistors were down around 5 to 7 percent of the input signal. The Transistor Section at Diamond Ordnance Laboratories has recently come out with some new circuits which I understand will be down around 2 or 3 percent noise.

We have also achieved this with tubes by properly mounting them and by using acoustic mismatch, having them in oil, and various approaches such as that.

J. Markowitz, ERA, Inc.: You mentioned that you had the problem not only with tubes but with other electronic components. And one possible solution was a low-frequency system. What do you mean by "low frequency?" What is the order of magnitude? Have you tried anything along those lines and how can this be accomplished?

Warren: We have started experimenting with it. We haven't gone far enough to have definitive results. In effect, in this system, the entire first stage in the amplifier is floated in oil. There are two things you have to worry about: metal-borne vibration and airborne vibration. Separating these is rather difficult, but once you have it in oil and there is acoustic mismatch, there is a big improvement over the airborne vibration. Then you only have to worry about what comes to it through the metal. In this case, it was arranged on a diaphragm which, because of its section modulus, would only vibrate in one plane, and as it vibrated it had to pump oil from one side of the diaphragm to the other. We haven't carried this far enough yet but the results look very promising. The noise levels were down around two or three percent, although we were using a vibration resistant tube in that case; one of the best tubes we could get our hands on. I believe that was a 6533.

Stern, NOL, White Oak: May I ask what is the frequency of the diaphragm system?

Warren: It was considerably higher than we desired because the diaphragm, in this case, consisted of plastic glass laminate. We aren't exactly sure of the natural frequency of that diaphragm because there was no sharp resonance. We feel, though, that it was between 50 and 100 cycles, and our next attempt will be at a much lower frequency.

Stern: Did you ever try using a rubberized hair base, or is it horse hair? It is a commercial material.

Warren: Yes. It appears to be very good from the standpoint of audio noise, but it takes up considerable space. Our approach at this time is only to do this for the worst offenders, such as the first stage of an amplifier. Then the remainder of the circle would be of high natural frequency construction, which takes much less space and is very easy to construct.

Stern: One other question. Would you give a brief description of the chamber you used? How do you control the pressure, and, if you do simulate high altitude, how you are able to cool or heat the air without using dry ice?

Warren: This chamber has a complete refrigeration system which goes with it. It is able to handle the load. The volume of the chamber is approximately five cubic feet, and it is completely sealed so we can pump the pressure down.

Stern: Do you have a rubber bellows? How do you seal it so you won't get leakage on the vibrator table?

Warren: The complete vibrator is placed in the sealed chamber. Then there is a diaphragm which is placed over the vibrator through which the table protrudes so that we draw down the pressure on the whole chamber. The temperature is concentrated in the top portion of the chamber so that the heat of the vibrator doesn't give us trouble.

R. M. Lambert, NOL, Corona: You mentioned that you had used variable inputs to the vibrator. I wondered whether you had done any work on complex wave testing.

Warren: We do not have a complex wave vibrator at Diamond Ordnance Fuze Laboratories, but we are very interested in following the work of other people on that subject. However, where we cannot exactly simulate a flight—this matter of flight simulation is always relative. It appears from the data we have that this simulation is not very close to reality. Regardless of what you do with this information, if it is not reliable to start with, it is not when you finish. The question is, then, what degree of unreliability do you want to accept? Do you want to go so far as the sine wave shake or do you want to use a complex vibrator? These are problems we are going to have to solve before we spend the money.

Lambert: At this time, would you venture any opinion as to the validity of going to complex wave vibration?

Warren: I will state categorically that you will not get flight simulation. You will get a complex wave shake but as I stated, when we put an accelerometer at the tube clip, we do not get a sine wave. It vibrates at whatever frequency it feels like vibrating, or resonates at whatever frequency it wishes, in whichever plane it wishes, and with whatever frequency or wave that it cares to have, and it is never a sine wave.

R. McKay, Shock and Vibration Research, Inc.: Referring to the last question, when you say that you put an accelerometer on this vibrating device, is the nature of the input excitation to the vibrating device a sine wave?

Warren: Yes. The table is.

McKay: Then I suggest to you, sir, that the use of the word "transmissibility" may be at times a little misleading. It seems to me from what you say that transmissibility, plus the old-time word "microphonics," would be more indicative. I think it is a question of frequency resonance of the system and by "system" I mean the actual accelerometer that you are using. It seems to me quite a good analytical approach could be worked out on this basis.

Warren: There is no question but that it is so with some of these, for example, the accelerometer which I cited at the beginning of the talk. However, where you have a complex structure and a single-degree-of-freedom shake, the analytical approach is very difficult.

* * *

ACCELERATION WAVEFORMS OF SOME COMMONLY USED MECHANICAL VIBRATION TABLES

Alexander Yorgiadis, Sandia Corp.

Experimental acceleration-time records of various types of mechanical vibration tables are presented. In addition to the fundamental sine wave of vibrations, there are superimposed high-frequency random harmonics which are of substantial magnitude. These undesirable harmonics were found to originate in gears and ball and roller bearings, or other regions of repeated localized impact.

INTRODUCTION

Vibration tables have been used for many years and have proved to be very valuable laboratory tools. It has been generally accepted over the years that vibration tables perform well as long as they execute motions of certain magnitude and frequency, and as long as they operate without failure of the table components. Little concern was shown in studying the actual waveform produced by the table, and its deviation from a simple harmonic motion.

When vibration tables were first developed, the science of instrumentation was not advanced to the point that would make waveform studies possible. Nowadays, however, a large variety of accelerometers and recorders are available that can supply reliable records of actual acceleration at the top of a vibration table. These records are very useful in evaluating and comparing different tables, and where necessary, can be of help in improving table performance. While this study is limited to mechanically excited tables, similar studies have been used in evaluating other types of vibration tables (References 1 and 2).

Conventional vibration tests are defined by specifying the frequency, direction, and amplitude of vibrations. At low frequencies, the amplitude is usually expressed in terms of inches of displacement or "double amplitude." At higher frequencies, it is customary to define amplitude in terms of g's of acceleration. In both instances it is assumed that the vibration table produces a pure, simple harmonic displacement and therefore a simple harmonic acceleration in the desired direction. It is also assumed that no vibrations exist in the two other orthogonal directions. Furthermore, all points on the table top are assumed to vibrate at the same amplitude and in the same phase at all times, the table acting as a perfectly rigid body.

It is generally known that these idealized conditions cannot be attained in practice. It is simply a question of minimizing the undesirable characteristics. In examining waveforms from some mechanical vibration tables, it was observed that in addition to the fundamental frequency of vibrations, there are superimposed high-frequency random harmonics which can be of substantial magnitude. Furthermore, these random harmonics are not limited to one direction of motion, but exist in all three directions, and even exist when the machines are operated without any table amplitude at the fundamental frequency. The magnitude of these harmonics varies along the surface of the table.

These harmonics are undesirable. They subject the specimen to test conditions which are not specified, and which do not probably exist in actual service. The test engineer usually ignores the presence of these harmonics, thus reducing the reliability of the tests, particularly in case of failure of the tested item.

TEST EQUIPMENT

A total of five different vibration tables were tested. The table identified as Table A was a positive-drive type table, while Tables B, C, D, and E were reaction type tables. All tables were tested with no specimens or weights attached to them in order to eliminate the specimen as a variable. It is expected that the addition of certain types of specimens to the tables would improve table performance, and other specimens would have the opposite effect.

Unbonded strain-gage accelerometers of 100-g capacity were used, and were mounted so as to measure the acceleration in three mutually perpendicular planes. The complete system, namely, accelerometer, amplifier, and recorder, was calibrated simultaneously and the over-all frequency response is plotted in Figure 1. This equipment is satisfactory for frequencies up to 600 cps, but the response falls off rapidly at higher frequencies, reaching 60 percent at 800 cps and 30 percent at 1,000 cps. Harmonics of frequency higher than 1,000 cps were observed in some tables, and these were detected by means of crystal accelerometers, which have a satisfactory response up to 2,000 cps and beyond.

TEST RESULTS

Records were obtained of acceleration versus time at three operating speeds, namely, 20, 40, and 60 cps on Tables A, C, D, and E. Table B was tested at 20, 40, and 55 cps. At each frequency, records were made at the minimum amplitude setting of the table and at an amplitude setting of the table necessary to produce an acceleration of 2.5 to 5 g. The actual accelerations used in each test are indicated on the records. Several of the records have appreciable high-frequency harmonic content, and the

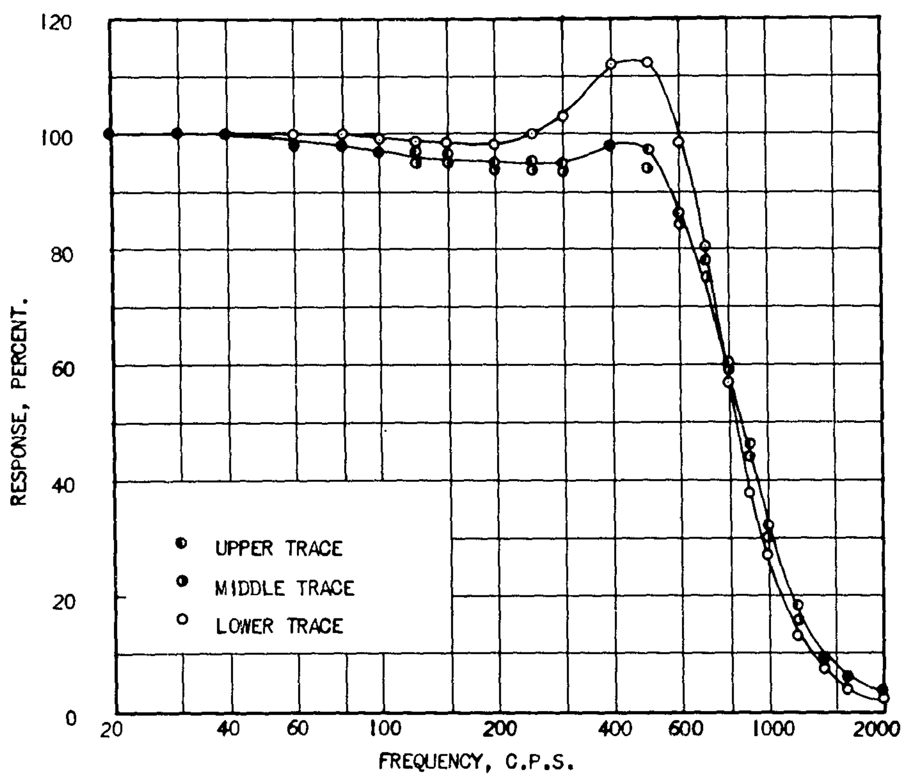

Figure 1 - Frequency response of strain gage accelerometers and recording systems as used in tests

magnitude of these harmonics can be evaluated by comparing their amplitude to that of the fundamental wave. No effort was made to tabulate quantitative values of harmonic content because these vary greatly from one location to another on the table and are subject to change with frequency and amplitude. Furthermore, the harmonic content is liable to change as test weights are added to the table. The results to be presented are therefore representative of the general behavior of the various tables tested, but cannot be used to predict the exact performance of these tables under conditions that are substantially different from those used in the tests.

Tests on Table A are presented in Figures 2, 3, and 4. Table A is a positive-drive table, vibrating in the horizontal direction. Such tables are also known as direct-drive tables. Table A has ball bearings on the main rotating shaft and sleeve bearings on the oscillating shaft. The machine is clamped to a heavy seismic concrete base. The harmonic content was relatively small at 20 cps, but became progressively greater at 40 and 60 cps. At 60 cps, these harmonics are estimated at 1 to 1.5 g. It was noted that the harmonics in the transverse direction are considerably smaller than those in the other two directions.

Recordings from reaction-type Table B are shown in Figures 5, 6, 7, and 8. This is a table for vibration tests in the vertical direction. It has roller bearings on the oscillator shafts, but no gears are mounted directly on the vibrating assembly. The gearing is located remotely, and universal joint shafts are used to drive the oscillator shafts. The bolted aluminum table structure is seismically supported and guided by means of rods with spherical rod-end bearings. In this table, very severe harmonics of approximately 5 g occurred at 60 cps at the center of the table.

The reaction-type vibration Table C is suitable for vibration tests either in the vertical or the horizontal direction. The oscillator shafts are supported on roller bearings with bronze retainers. The synchronizing gears are mounted into the vibrating table structure, which is a steel weldment. This table had a large amount of harmonic content at all operating conditions, and the frequencies of these harmonics were in the range of 600 to 2,400 cps. Both crystal and strain-gage accelerometers were used; they were mounted at two different points on the table, namely, in the middle of a panel on the table and immediately over the intersection of two stiffeners.

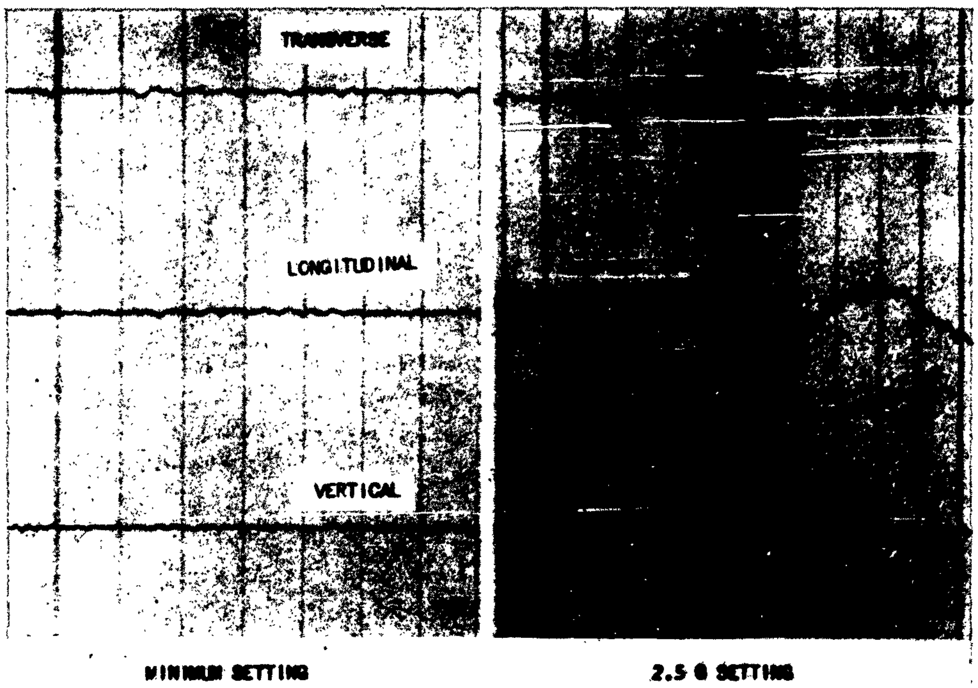

Figure 2 - Acceleration-time records on positive-drive vibration Table A at 20 cps with strain gage accelerometers

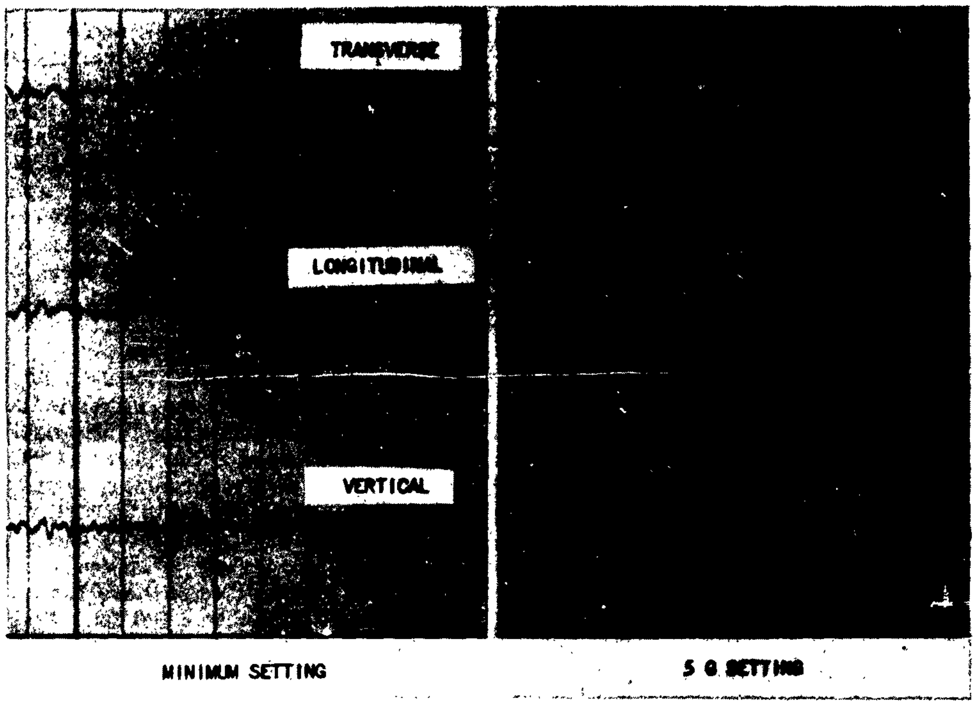

Figure 3 - Acceleration-time records on positive-drive vibration Table A at 40 cps with strain gage accelerometers

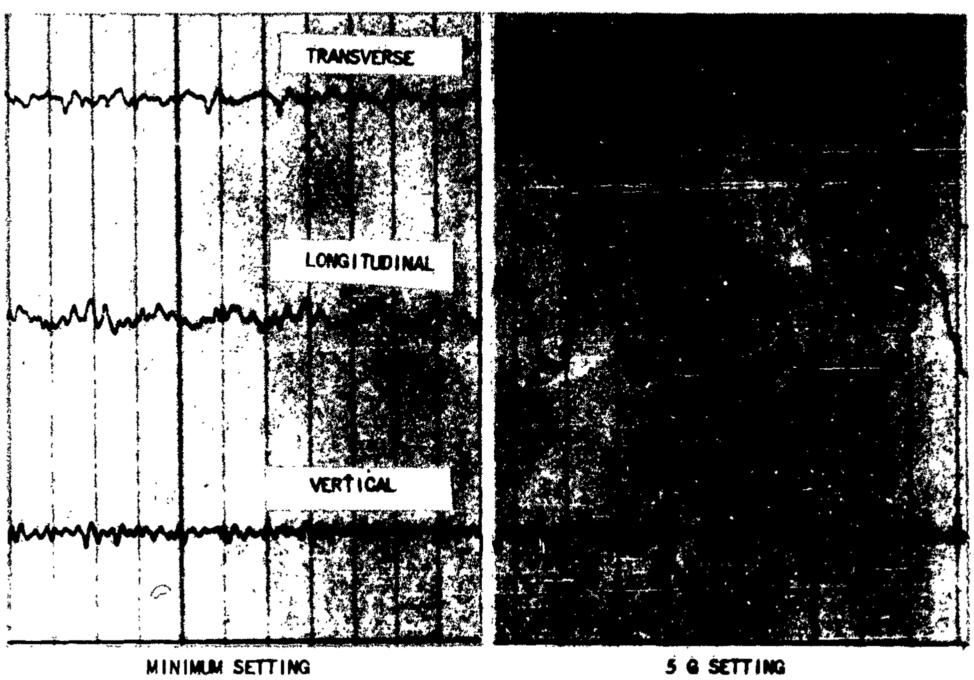

Figure 4 - Acceleration-time records on positive-drive vibration Table A at 60 cps with strain gage accelerometers

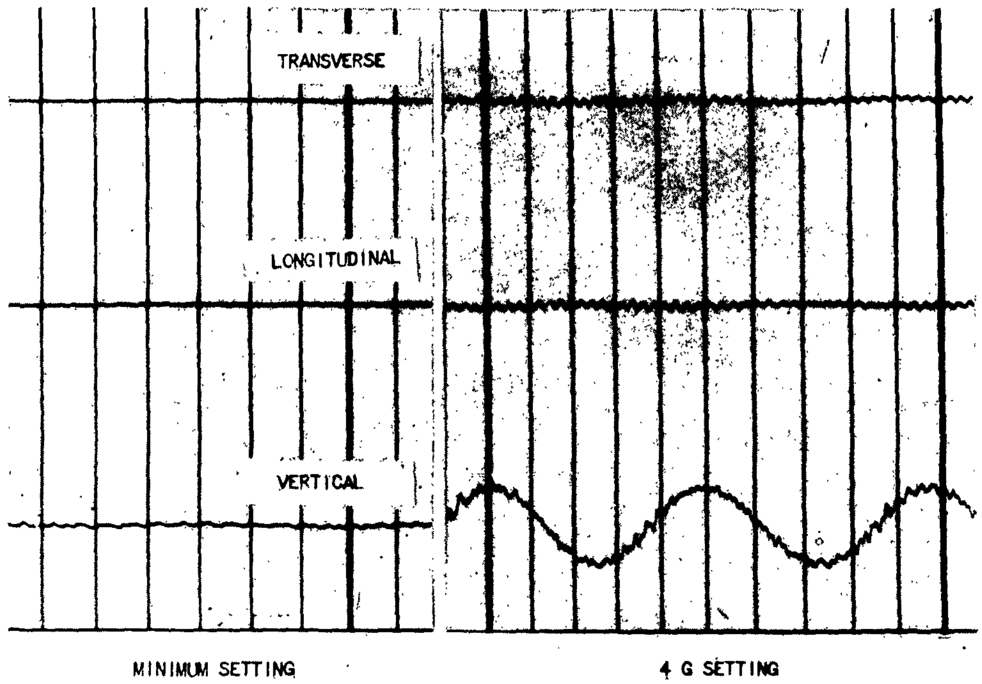

Figure 5 - Acceleration-time records on reaction-type vibration Table B at 20 cps with strain gage accelerometers located to one side of table top

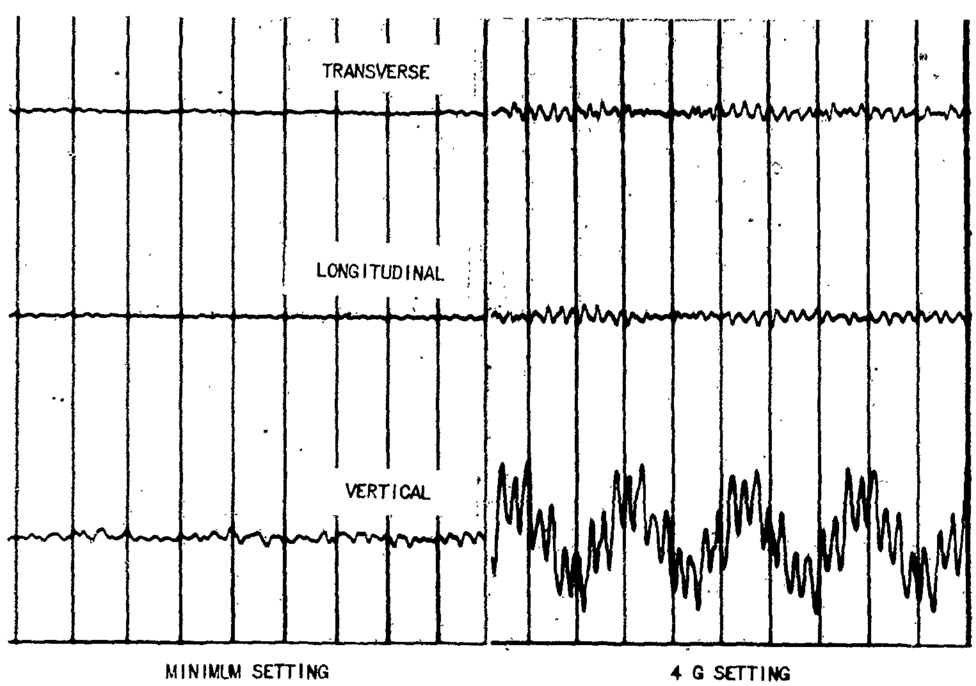

Figure 6 - Acceleration-time records on reaction-type vibration Table B at 40 cps with strain gage accelerometers located to one side of table top

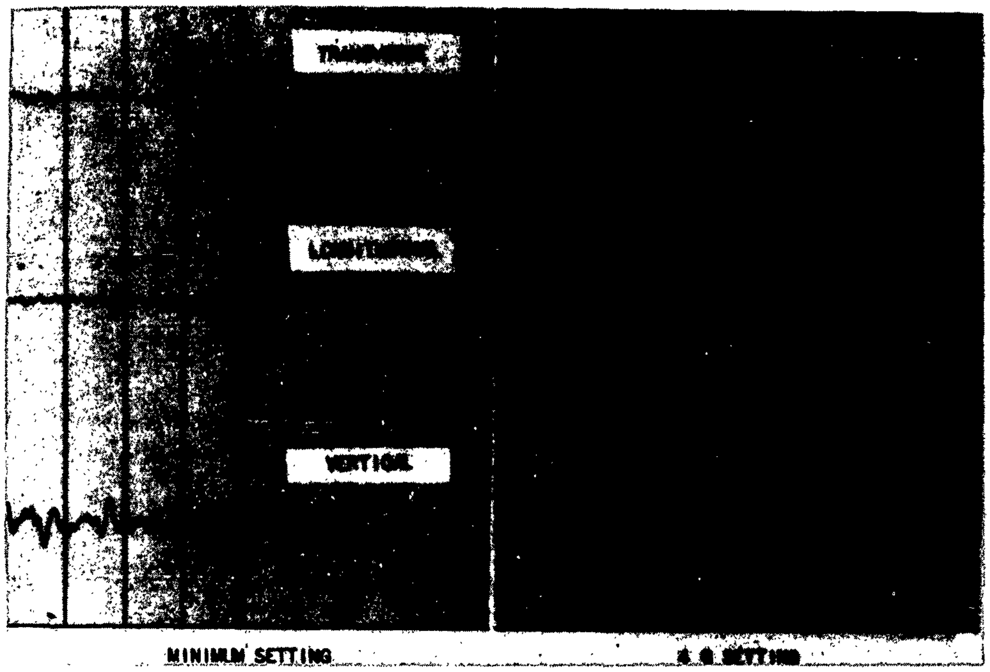

Figure 7 - Acceleration-time records on reaction-type vibration Table B at 55 cps with strain gage accelerometers located to one side of table top

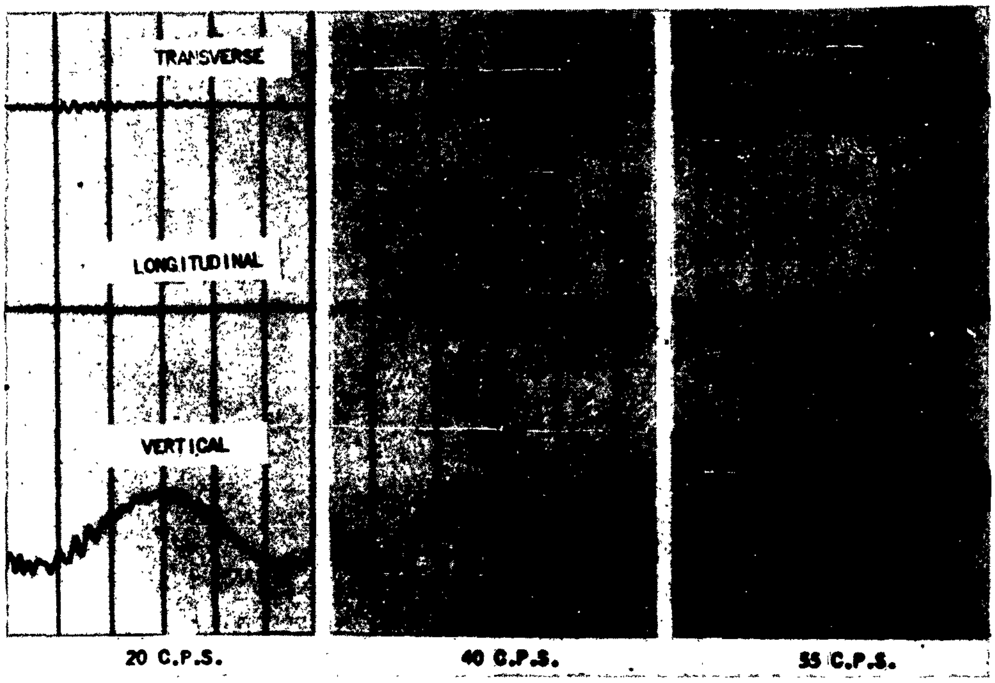

Figure 8 - Acceleration-time records at 4 g setting on reaction-type vibration Table B with strain gage accelerometers at the center of table top

In order to insure that the observed harmonics were actually produced by mechanical motion of the table and not as a result of other effects, additional accelerometers were mounted on a rubber pad on top of the table. This resilient support was designed to have a natural frequency of 150 cps, so that the fundamental vibrations up to 60 cps could be transmitted without substantial change, but frequencies in excess of 600 cps were isolated.

Records from Table C are presented in Figures 9, 10, and 11. As expected, accelerometer outputs from the units on the rubber pads show almost no high-frequency harmonics. The accelerometers over the stiffeners show appreciable harmonic content, and those over the mid-panel contain still higher harmonics. Peak harmonic accelerations as high as 6 g in magnitude were observed.

One peculiarity of this table was that the harmonic intensity at any given speed of operation was nearly the same at all vibration settings. The strain-gage accelerometers could not indicate the harmonics occurring at 40 and 60 cps since their frequency was very high.

Reaction-type vibration Table D is a table similar but smaller than Table C, in which the table structure is made of bolted aluminum members. This table is an improvement over Table C. Only strain-gage accelerometers were used (Figures 12, 13 and 14), and the higher harmonics were attenuated or filtered. For example, at 20 cps operating frequency, the harmonic frequency is about 1,000 cps, and the apparent amplitude is only 12 percent of the fundamental. However, knowing that the frequency response at 1,000 cps is only 30 percent, the harmonic acceleration is actually 40 percent of the fundamental, or, roughly 2 g.

Accelerometer recordings from vibration Table E are shown in Figures 15 to 18 inclusive. This is a horizontally vibrating table, using sleeve bearings to support the oscillator shafts. The gearbox is supported remotely, and the oscillator shafts are driven through rubber couplings. The vibrating assembly is a steel weldment guided by means of metal flexures. A heavy seismically supported base absorbs the reaction moment and stabilizes the vibrating assembly. The records obtained on this table indicate that it is practically free of high-frequency

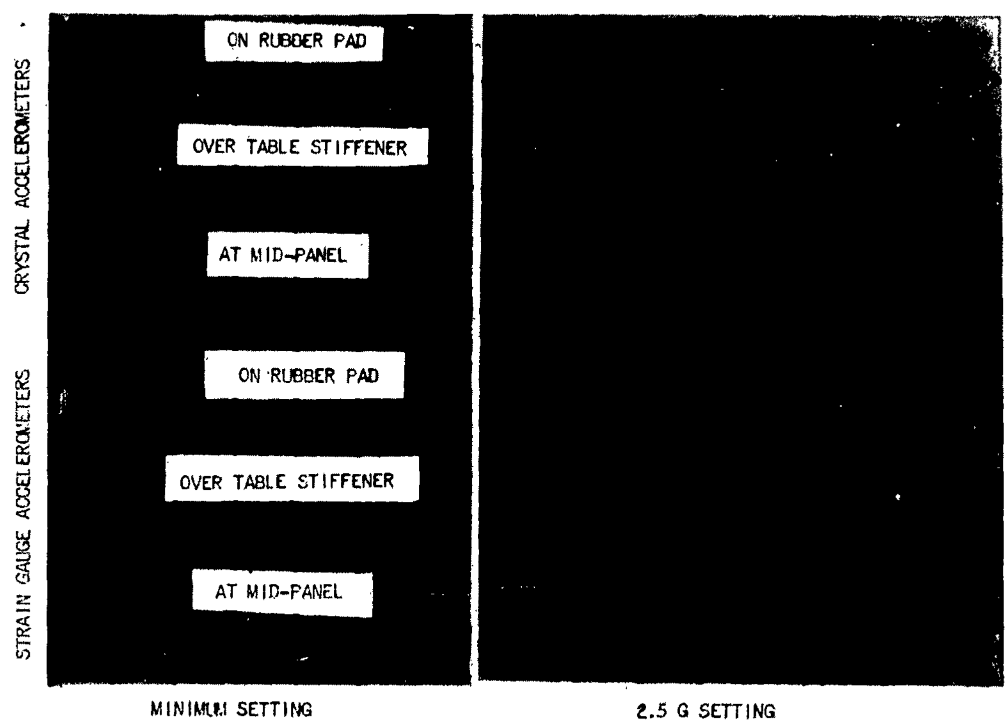

Figure 9 - Acceleration-time records on reaction-type vibration Table C at 20 cps in direction of required vibration

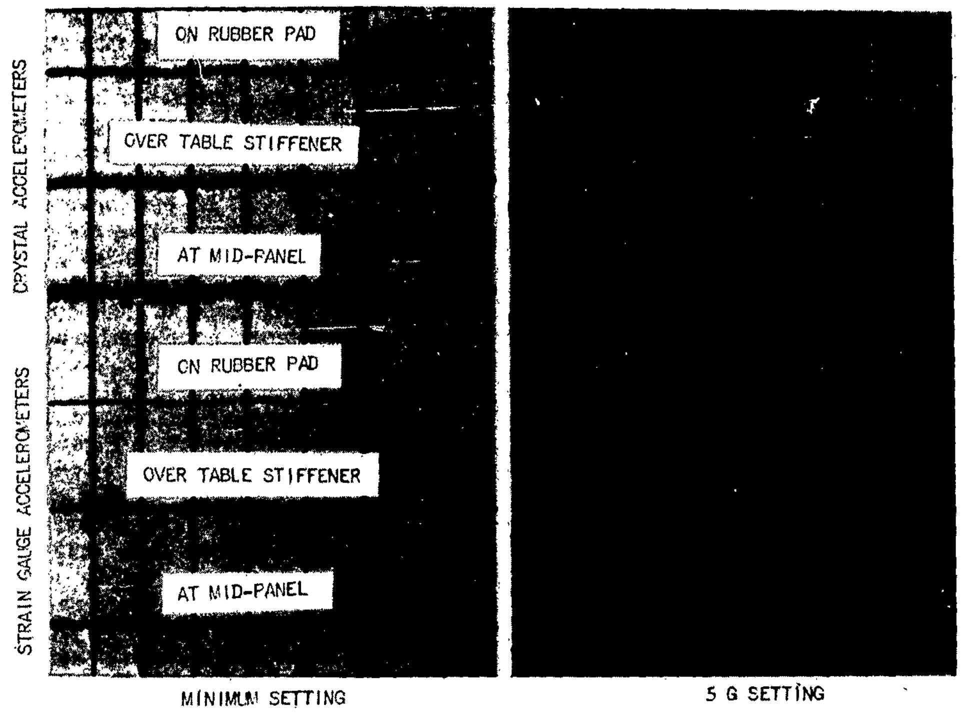

Figure 10 - Acceleration-time records on reaction-type vibration Table C at 40 cps in direction of required vibration

harmonics. Furthermore, no vibrations were detected in directions other than those required. Figure 18 shows some records obtained with crystal accelerometers mounted in the same manner as those used with Table C. Even sensitive crystal accelerometers could not detect appreciable high-frequency harmonics in this table.

CONCLUSIONS

1. Most mechanical vibration tables generate high-frequency harmonic vibrations in addition to the desired fundamental mode of vibrations.

2. These high-frequency harmonics can be measured with strain-gage type accelerometers or crystal accelerometers.

3. These high-frequency harmonics can exist in all three planes even though the table is intended to vibrate only in one direction.

4. In general, the intensity of these harmonics increases with operating frequency.

5. The harmonic content is usually greater at the middle of panels on the table than over the table stiffeners.

6. Tables which include gears and ball or roller bearings in the vibrating structure produce a substantial amount of these undesirable harmonics. Oscillators using only sleeve bearings and no gears are practically free of such harmonics.

7. The high-frequency harmonics appear to be caused by localized impact in such parts as balls, rollers, or retainers in antifriction bearings and in meshing gear teeth. The harmonics are amplified as a result of "ringing" of panels on the table structures, which usually have very low

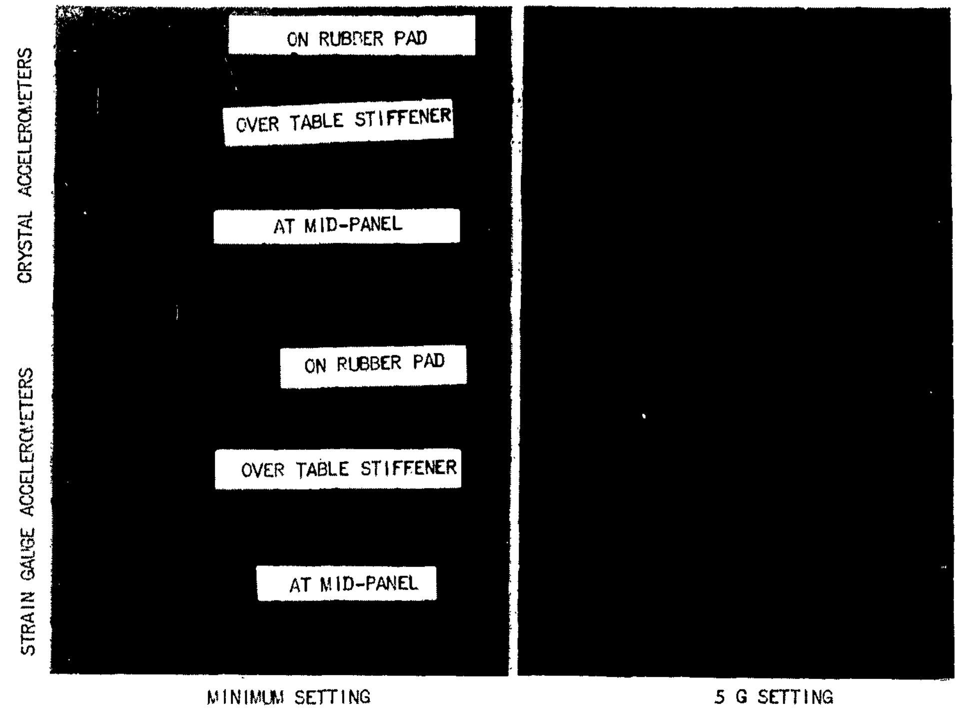

Figure 11 - Acceleration-time records on reaction-type vibration Table C at 60 cps in direction of required vibration

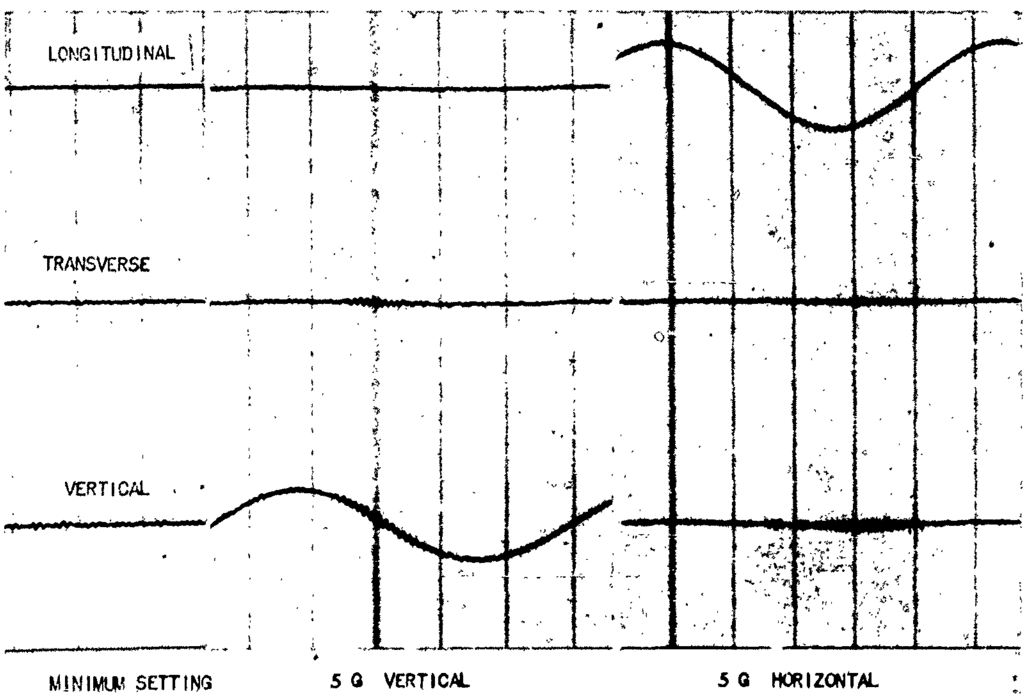

Figure 12 - Acceleration-time records on reaction-type vibration Table D at 20 cps with strain gage accelerometers

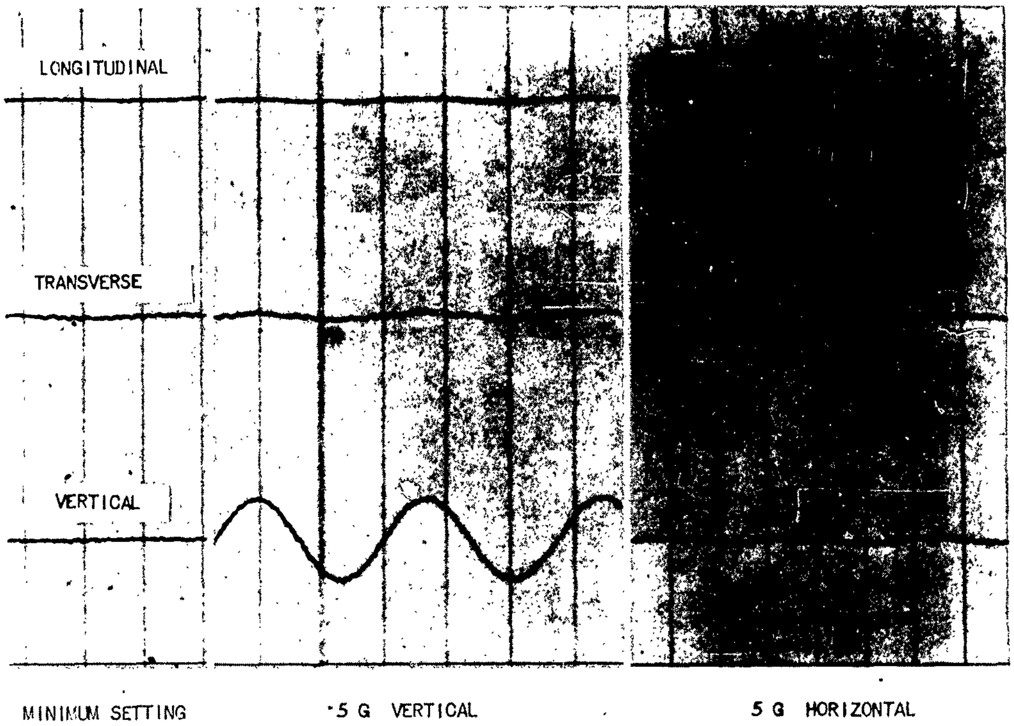

Figure 13 - Acceleration-time records on reaction-type vibration Table D at 40 cps with strain gage accelerometers

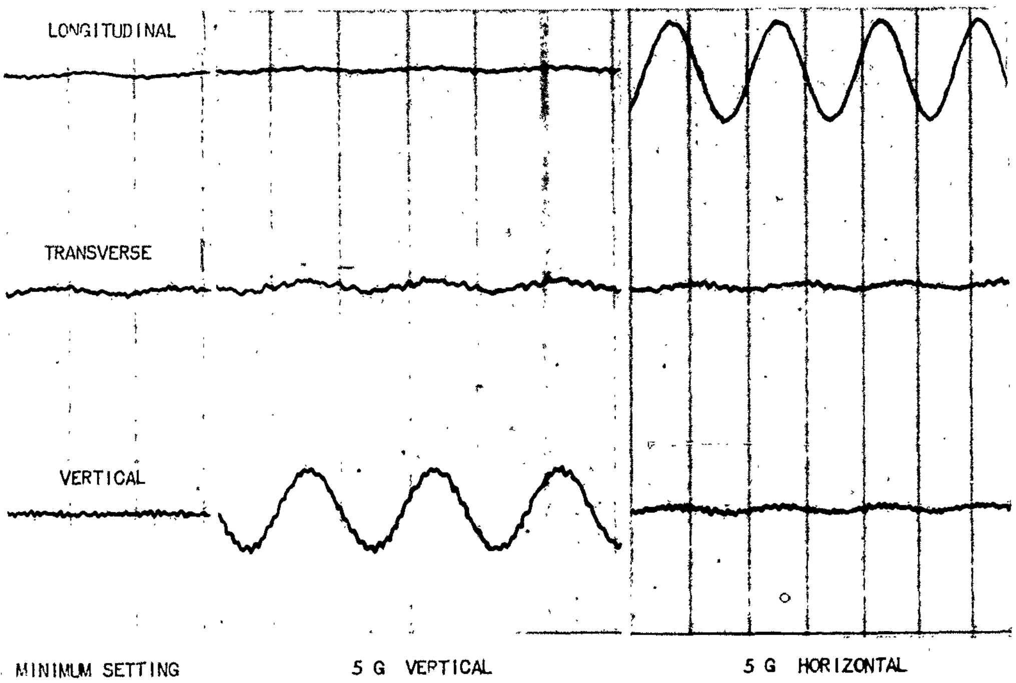

Figure 14 - Acceleration-time records on reaction-type vibration Table D at 60 cps with strain gage accelerometers

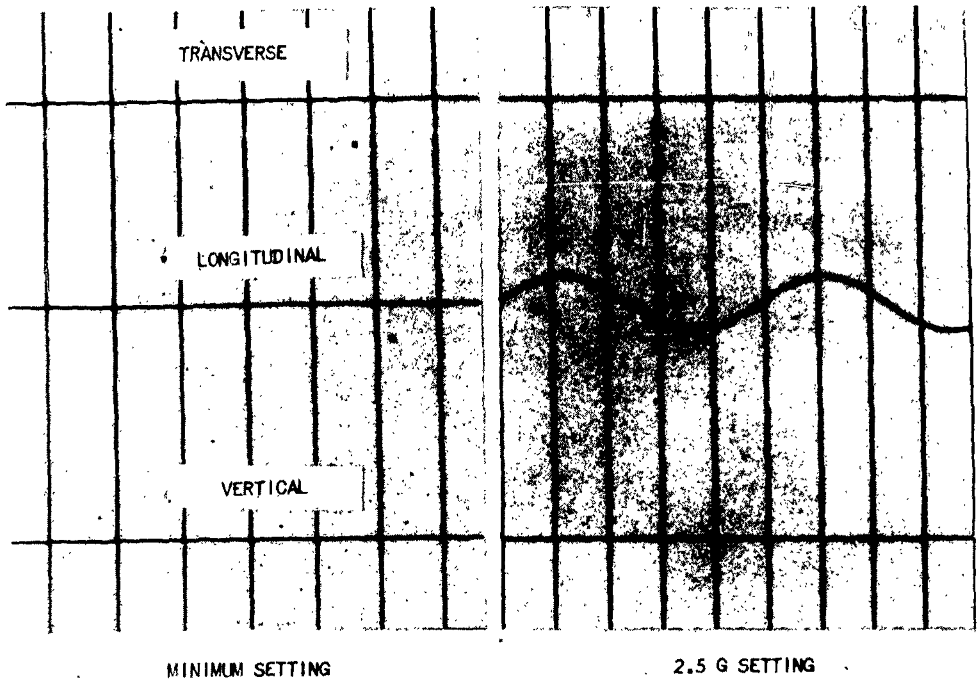

Figure 15 - Acceleration-time records on reaction-type vibration Table E at 20 cps with strain gage accelerometers

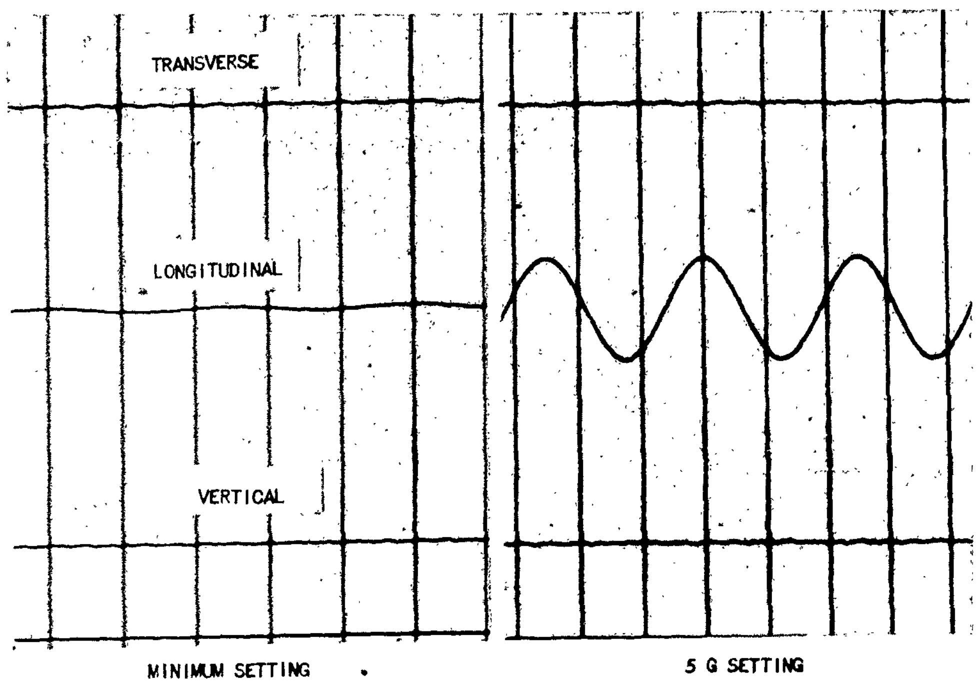

Figure 16 - Acceleration-time records on reaction-type vibration Table E at 40 cps with strain gage accelerometers

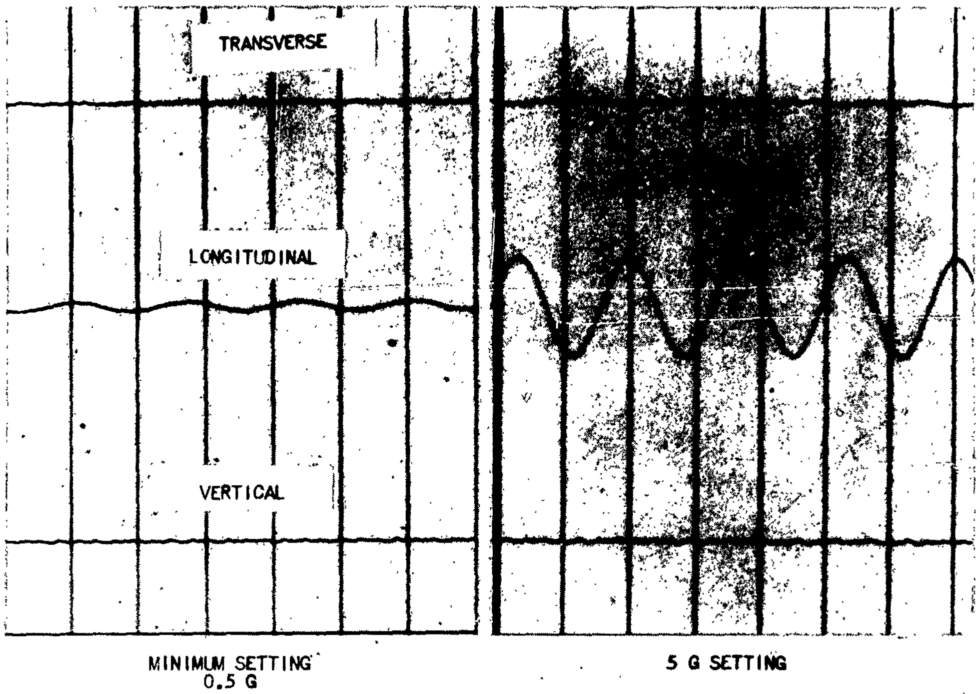

Figure 17 - Acceleration-time records on reaction-type vibration Table E at 60 cps with strain gage accelerometers

damping properties. The film of oil in sleeve bearings acts as a cushion and does not produce such high-frequency harmonics.

ACKNOWLEDGMENTS

The author wishes to thank C. N. Giles and R. S. Jacobson for their valuable assistance on the instrumentation and for obtaining the records.

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DISCUSSION

Seely, NOL, White Oak: On these harmonics, I submit that they are not always undesirable. For example when you have testing which can only be done on mechanical vibrators, these harmonics may well be a test of the large objects at the higher frequencies. The problem lies not in their existence but in lack of control.

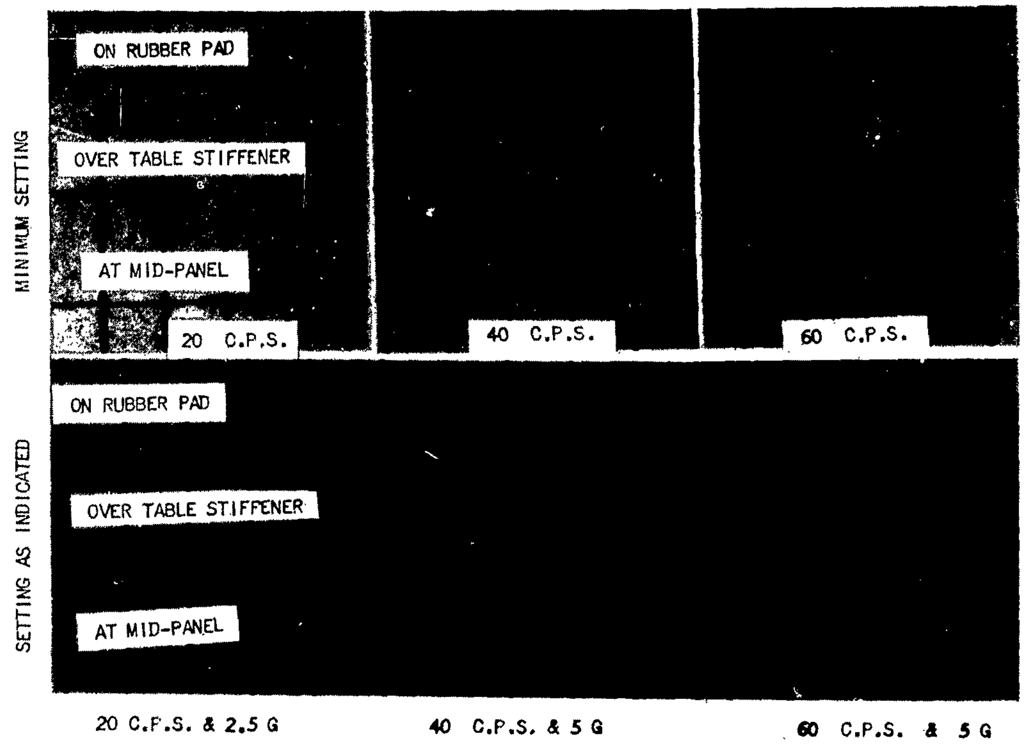

Figure 18 - Acceleration-time records on reaction-type vibration Table E with crystal accelerometers

Yorgiadis: Something which is not specified in the test is usually something you do not want. If you have to test to 1 g at low frequency and if, in addition, you have 5 g's of high frequency which you cannot remove during your test, then it seems to me that that is something you really do not desire. On the other hand, I can see that if you want both high and low frequencies mixed together in a test, and if you could adjust both of these to the values you want, that would be desirable. However, in the tests where I have experienced these high frequencies, they were not something that was wanted and if the choice had been available to us we would have eliminated or minimized them.

Sanders, Bendix Aviation: Have you run any tests of a similar nature on electrodynamic shakers?

Yorgiadis: We have run some tests but did not include them in this paper. Of course the electrodynamic shaker does not have any gears or bearings, so when you get distortion it is due to other effects, mostly at high frequencies and approaching the limits of the equipment in the sense that localized resonance occurs in the machine or specimen. Any distortion in the sine wave vibration is usually small and the majority of the frequencies will be amplified even if one of the harmonics in the exciting signal coincides with the natural frequency of a particular part of the table. The result might be to have third and second harmonics mixed up at certain points on the table, but the nature of these harmonics, at least in our tests, did not anywhere near approach the nature of the records we have just seen.

Seaman, BuShips: The speaker raises a very good point on the high-frequency problem in most of these mechanical shakers. I think that some thought might be given to the incorporation of mechanical filtering, such as a high-frequency mounting, in order to eliminate some of this high frequency, particularly if you do not want it on the test equipment. We can't scrap all our mechanical shakers.

Yorgiadis: That is precisely what we were trying to show when we mounted the accelerometer

on rubber pads. It does not seem to be difficult to isolate frequencies over about 600 cps on heavy stuff. I think any type of isolating device would eliminate them and yet would transmit without much amplification the frequencies between, say, 10 and 60 cps. I believe that anybody concerned with this high frequency, who has an idea it might be doing some damage other than desired, should have no difficulty at all in mounting the test object on a seismic mount.

* * *

THE NEED TO CONTROL THE OUTPUT IMPEDANCE OF VIBRATION AND SHOCK MACHINES

Machines for simulating shock, sinusoidal vibration, and random vibration are usually designed to duplicate the envelope of typical field motions regardless of the mechanical impedance of the item under test. This neglect of the effects of impedance generally leads to tests of real equipment which are far more severe than any field conditions.

R. E. Blake, NRL

The purpose of this paper is to stimulate more interest in the problem of simulating the output impedance of equipment foundations in vibration or shock tests. There is a superficial appearance that this is not a problem today because of the fact that output impedance is seldom mentioned in reports of current work. However, it is probably more correct to say that the people who have thought about the problem have decided that it is complex, that a great deal of data is required to do anything about it in a practical way, and that present machines err on the conservative side.

This is a sound interim solution of the problem as it shapes up to accomplish the practical test work of the day. Unfortunately interim solutions tend to perpetuate themselves unless the shortcomings of these stopgaps are kept in mind. Therefore, in spite of the fact that this paper does not solve the problem, it serves the useful purpose of bringing the flaws of our present practice into sharper focus.

The output impedance of a vibration table is the measure of its ability to maintain nearly-constant table amplitude in spite of the reaction forces of the equipment being vibrated. Output impedance is the ratio of the force exerted on the table to the change in table vibrational velocity. In general, impedance is a function of frequency. It can be measured easily by observing the effect of adding a known mass to a vibrating table or the effect of applying sinusoidal force to the table.

A direct-drive vibration machine has a high output impedance because it maintains a preset amplitude even when resonance of equipment under test produces a high inertia reaction. The electromagnetic machines or the ones shaken by rotating eccentric weights do not have an inherently high impedance. But if the amplitude falls off due to resonance of the test structure, it may be restored manually, or by automatic feed-back control, in many machines.

Now every foundation for equipment in a ship, plane, rocket, or anything else which vibrates has its own characteristic impedance as a function of frequency. Unfortunately, it has not been customary in the past to measure impedance, although we have made many measurements of foundation vibration amplitude as a function of frequency. If one measured the vibration in an airplane, for example, he might obtain the lower curve sketched on Figure 1. The amplitude executes a series of ups and downs as a function

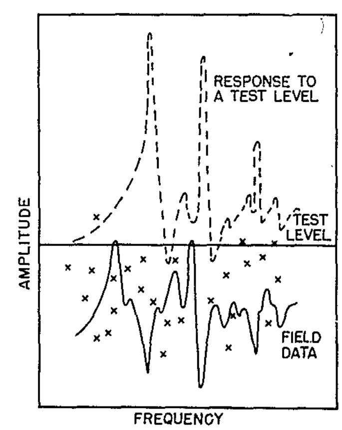

Figure 1 - Vibration amplitude-frequency curves

of frequency; the high points are at several times the level of the low points.

This sort of data has been gathered from several locations in a great many aircraft and it is the usual practice to plot all the data as a "flyspeck plot" on one sheet in the manner shown on the graph. The next step is to draw a smooth envelope which is higher than practically all of the points and to use this as the specification for laboratory tests of aircraft equipment. Usually a piece of equipment must be tested for possible service in any one of a number of airplanes, or in some airplane which has not yet flown. Very rarely do we know the actual vibration curve which the equipment will experience in service.

Suppose that the equipment which was on the foundation when this lowest curve was measured were to be subjected to the laboratory vibration test. The amplitude at some point in the equipment itself can be expected to vary in some fashion like the curve marked "Response." There will be resonant frequencies at which the equipment amplitude becomes very high, and danger of failure is greatest. At these resonant frequencies the inertia reaction against the table is also a maximum; in fact, a resonating mode of vibration of an equipment acts just like the vibration absorber which is discussed in most texts on vibration. Unless the output impedance of a foundation is very high, the vibration absorber effect produces a pronounced dip in the foundation amplitude. This has been indicated by showing low foundation amplitudes at the resonant frequencies of the equipment. Here is the thing about the laboratory test which is disturbing. At the frequencies where damage is most likely, the table amplitude is kept high even though the actual foundation amplitude would drop to a much lower value.

The laboratory test level was not determined by the foundation amplitude at the resonant frequencies of the equipment but by the peaks of foundation amplitude. Except for the possibility that the force causing the vibration reached a peak of foundation amplitude can be attributed to a resonant frequency of the combined airplane-equipment structure. But the outstanding characteristic of a resonance condition is that only a small force can produce a large amplitude; in other words the impedance of the foundation (with the equipment in place) is a minimum at this frequency. This infers that any change in the equipment which changes its inertia reaction would have a pronounced effect on the foundation amplitude at this frequency. This apparent "softness" of the resonant peaks is perhaps a little misleading.

If the response in the equipment to the foundation motion shown in Figure 1 were plotted we would find that the peaks of equipment response coincided in frequency with the peaks of foundation amplitude. But the ratio of equipment amplitude to foundation amplitude at these peaks would be much smaller than was found in the laboratory test because the frequency is different. Thus it appears that our laboratory test machines are often much more violent than the foundations which are to be simulated. This is due to our failure to duplicate the effect of foundation impedance.

It can be argued, correctly, that exceptions to this dip in foundation amplitude will occur if the foundation impedance is very high. Since the impedance to be encountered in service can seldom be known when a laboratory test is run, the safest thing to do is to use an infinite table impedance—as we do now. I see no alternative at present to the practice, which Dr. Morrow described in an earlier paper as "filling in the valleys," at least so far as vibration work is concerned. Nevertheless, our present practice is not very satisfactory, since we can show that our tests are much more severe than might be necessary if we understood the effects of impedance better. We should not give up without a struggle; and it seems that the struggle to simulate the effects of output impedance is yet to be made.

So far, the discussion has referred only to sinusoidal vibrations, but the arguments apply as well to the random vibrations which have become important in missile work. The transmission of random vibration in a structure is directly related to the behavior under sinusoidal vibration, as Mr. Crede pointed out in his paper, and the effects of foundation impedance are very much the same. When we plot spectral density curves from a variety of rockets on one graph and draw an envelope, we are just as likely to be overconservative as we were in drawing an envelope for sinusoidal vibrations.

This discussion of vibration output impedance is based only on theory. There has not been any experimental work as yet; but it has only been argued that here is a subject deserving further exploration both in theory and experiment. This study of vibration impedance was actually a by-product of a very similar problem which arose from our work on shock in submarines.

The theory of the effect of output impedance on shock has not been completely worked out; in fact there isn't even a word for an impedance effect in the parlance of shock. This looked to us like an opportunity to coin a resounding new word to add to the vocabulary of shock, but we came back to using output impedance itself because of a hunch that sooner or later a student of Fourier transforms will show that vibration and shock impedance are fundamentally the same thing.

Although shock output impedance cannot be defined or explained exactly, there is some experimental evidence that it is real. This evidence appeared in the shock spectra measured on the foundations of some rather heavy equipments which were installed in submarines for underwater explosion tests. A distinct dip in spectrum amplitude was found to coincide with natural frequencies of the equipment. In order to show why this was so interesting to us, it is necessary to fill in some background.

Most of our understanding of what happens to the structure of a piece of equipment when its foundation goes through a shock motion (or a vibration) is contained in the normal mode theory of shock response of a multi-degree-of-freedom structure. Equation (1) is a result of this theory.*

$$\sigma_i(t) = \sum_{n=1}^{f} \left[\frac{\sigma_{in} \int_M \phi_n \, dm}{\int_M \phi_n^2 \, dm} \right] \frac{1}{\omega_n} \int_0^t \ddot{x}_o \sin\omega_n(t-\tau)d\tau. \quad (1)$$

It gives the stress, σ, at some point i in the structure as a function of time, t. Fortunately it is not necessary to go into too much detail about this rather complex looking expression except to say that it is a series of terms; there is one term for each normal mode and natural frequency ω_n of the structure. The expression within the brackets is determined entirely by the characteristics of the structure; the mass distribution, m, the normal mode shape, ϕ_n, and the stress, σ_{in}, which accompanies vibration in the mode shape, ϕ_n.

The remaining portion of each term is a function of the acceleration, $\ddot{x}_o$, of the equipment foundation and the natural frequency, ω_n, of the mode. So in order to be able to calculate stress in an equipment subject to a motion of its foundation, we must know the value of these integrals for the natural frequencies of the equipment. The integrals could be evaluated by some form of analog or computer, but the most direct method is to take advantage of the fact that the equation for the deflection d of a very simple mass-spring structure is a simplified version of the equation just shown. In fact, d (Equation (2)), is identical to the quantity which we wanted to evaluate.

$$d = \left[\frac{1}{\omega_n} \int_0^t \ddot{x}_o \sin \omega_n(t - \tau)d\tau \right]_{max}. \quad (2)$$

We can place an instrument called a reed gage, or shock spectrum recorder, beside the equipment on its foundation, and measure the deflection, d, of the different mass-spring elements in the gage. These deflections are a function of natural frequency and give us the data needed to compute stress in the equipment structure. A plot to maximum reed response, d, as a function of natural frequency is a "shock spectrum."

During recent underwater explosion tests of submarines in Chesapeake Bay, shock spectra were measured at several equipment foundations. Then the spectra were divided into classes in which each class represented nominally similar equipment locations. For example, Figure 2 is a set of spectra obtained for a class defined as "shock motion normal to the hull for foundations attached directly to the hull and supporting equipment weighing 1,000 to 8,000 pounds."

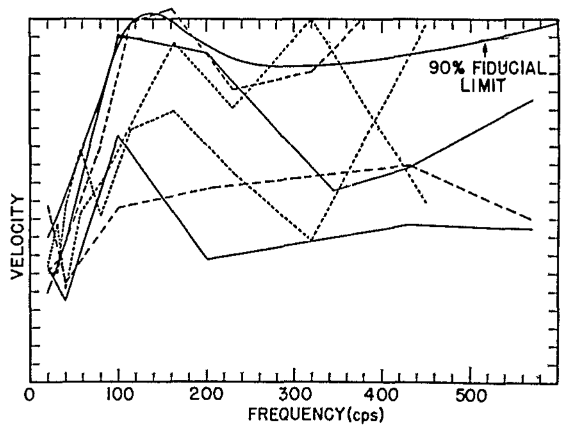

Figure 2 - Shock spectra from similar locations in a ship

These spectra formed a loosely correlated group which showed significant differences in level from classes based on other criteria. But one should not expect there to be any precise

*R. E. Blake, and E. S. Swick, "Dynamics of Linear Elastic Structures," NRL Report 4420 of 7 Oct 1954.

agreement among the spectra of a class—and there was not. Next, to define a conservative shock spectrum for laboratory tests, or for design, we drew in a 90 percent fiducial limit curve, which is a smooth curve lying above 90 percent of the data points.

When we tried to design equipment to withstand this 90 percent limit, however, we ran into a real difficulty. The level was very severe and it was possible to prove that very few practical shapes of mild steel structure could be designed to undergo the 90 percent fiducial limit shock without severe permanent distortion. It was therefore a real mystery that so many mild steel structures came through the shock test with no apparent damage. We could think of many partial explanations but none were found adequate to close the gap until we started calculating natural frequencies of some of the equipments which had been on the foundations.

Figure 3 shows the spectrum at the foundation of a very heavy equipment mockup which was essentially a single-degree-of-freedom system. The fundamental natural frequency, ω_n, of that structure was calculated and was found to coincide with the abrupt dip in the spectrum. A strain gage on the structure enabled us to use the equation for stress (Equation (1)) to compute the shock spectrum value needed to predict the stress that was measured. The calculation yielded the point shown, which confirmed the frequency calculation in showing that the only spectrum value which had real meaning for this equipment was relatively low.

There is not too much additional data to confirm this shock-output impedance-effect because we did not set out to study it when the experiments were planned. From the few other cases for which calculations could be made, two more will be shown. Figure 4 is the spectrum at the foundation of the same equipment as Figure 3 except that Figure 4 is for athwartships shock motion. Again there is a dip in the spectrum and pretty good agreement with the calculated point.

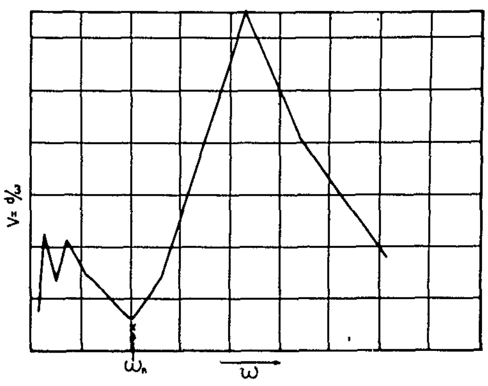

Figure 3 - Vertical shock spectrum at foundation of 37,000-lb structure

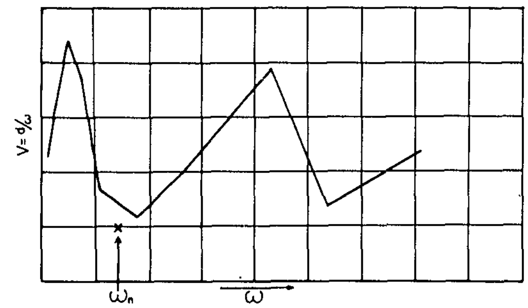

Figure 4 - Athwartship shock spectrum of foundation of 37,000-lb structure

The last figure, number 5, is for an essentially two-degree-of-freedom structure. Here we see the two natural frequencies and the calculated points coinciding with two low points of the spectrum.

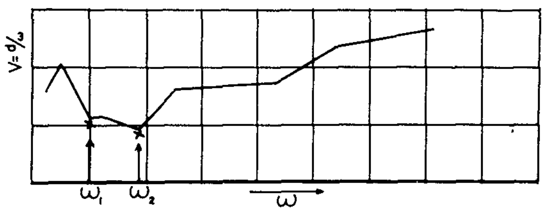

Figure 5 - Vertical shock spectrum at foundation of 31,000-lb structure

These data show that the inertia reaction of an equipment is quite selective and subtle in its effect on the foundation shock motion. The effect is pronounced only for the shock spectrum values which apply to the equipment itself, namely those for the natural frequencies of the equipment. When the 90 percent fiducial limit curves were computed, we used spectrum values whether they corresponded to equipment resonances or not. The high points which actually determined the 90 percent limit are therefore not at all representative of the spectrum values appropriate to the design and test of heavy equipment.

In simulating shock on submarines, the need to duplicate the output impedance of foundations is more pressing than in simulating vibration. The envelope of measurements which neglects the valleys of shock spectra promises a test which will be very hard to pass. In this case the factor of safety would be too big to live with, so we must find a way to cut down the factor without also reducing the safety.

In concluding it must be admitted that more questions were brought up than were answered. It is too early yet to say how much progress we can make toward understanding and simulating the output impedance of foundations. After all, nobody has yet made any measurements to show the impedance curves of typical foundations.

* * *

DISCUSSION

C. E. Crede, Barry Controls: The purpose of Mr. Blake's paper was to bring this problem into sharper focus and I feel sure that the subject matter is one that is going to receive a great deal more attention as the science of environmental simulation becomes more sophisticated.

Mr. Blake brought out one point that I think should be emphasized a little bit more because it may suggest something to somebody that might throw a little more light on the subject.

In the first slide a curve was shown of the environment as measured in aircraft, a plot of the amplitude versus the frequency, and it was pointed out that if you change the impedance of the structure or of the equipment that that structure supports, it would drastically change the shape of the curve. This indicates that most of these measurements were probably made on structures which supported equipment whose impedance itself was not known.

Now if the technique of taking environmental measurements could be modified, say, by making measurements only on structures which support a body of known impedance and then perhaps substituting another body of known impedance, we would thereby introduce a couple of knowns where unknowns now exist and some progress might be made toward calculating the impedance of the structure.

Blake: Thank you. One method of doing this was to install a weight of known mass and measure the vibration amplitude, then change that mass and measure the change in amplitude of vibration. You don't have to bring in a shaker and attach it to the foundation provided the foundation is already vibrating.

* * *

SHOCK-SPECTRUM CHARACTERISTICS OF THE NAVY MEDIUM-WEIGHT SHOCK MACHINE

Arthur F. Dick, NRL

This paper indicates the manner in which the shock responses of the machine may be derived. Upper and lower bounds of shock spectra that would be expected on the test load are presented, and some experimentally obtained reed-gage spectrum points are superimposed on the derived spectra.

During the past 15 years or so laboratory simulation of shock has been used more and more by the Navy in acceptance tests of equipments intended for submarines, surface ships, and aircraft. And most of this shock simulation has centered around the U. S. Navy Light-Weight High-Impact Shock Machine and the Medium-Weight Machine.

At last count there were 42 Light-Weight and 8 Medium-Weight Shock Machine locations, and so both types have been rather extensively used. They have also been calibrated, modified, standardized, talked about, and written about so that most everyone here has been exposed to them at some time or other. Those of you who are more intimately familiar with them know that the Light-Weight Machine—even though the shock motions produced on it may be empirically suited for production testing—cannot be very closely approximated by any simple system of masses and springs. For this reason, not much has been done about theoretical results of shock tests conducted on the machine.

On the other hand, the Medium-Weight Machine—as it is normally used in specification testing—can be well represented by two lumped masses and a spring. It is, then, not too difficult to derive expressions for shock-induced displacement, velocity, and acceleration of both masses, as well as the response of a single-degree-of-freedom reed on the load mass. It is the purpose of this paper to indicate the manner in which the shock responses can be derived, to present upper and lower bounds of shock spectra that would be expected on the test load, and to superpose on the derived spectra some experimentally obtained reed-gage spectrum points.

In this schematic model of the shock machine (Figure 1), the test load, M_1, is attached by standardized support channels of stiffness, k, to the anvil table M_2. When it is at rest, the anvil table—which weighs about 4,500 lb—is positioned a nominal table-travel distance of 3 or 1-1/2 in below a set of upper table stops. Shock is initiated by a 3,000-lb hammer, dropped from a specified height and hitting a hardened striking pad on the underside of the anvil table. At that time, the anvil table acquires an upward velocity so suddenly—in about a millisecond—that it can be considered an instantaneous velocity step, and the whole system then executes an upward translation in conjunction with an internal free vibration of the two masses and the spring. This combination of translation and vibration continues until the anvil table strikes the upper stops.

During this short interval of time, damping can be neglected and the only significant forces

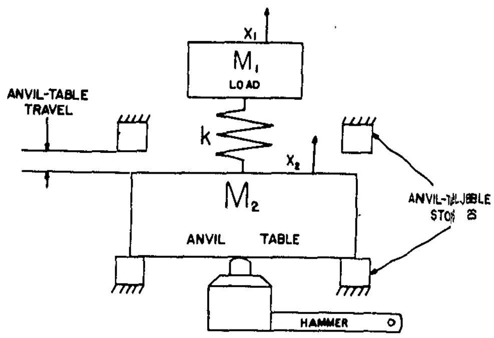

Figure 1 - Schematic of Navy medium-weight shock machine

acting on the system are those due to inertia, gravity, and spring deflection. Solution of the differential equations for these forces yields expressions for displacement, velocity, and acceleration of both M_1 and M_2. Now the maximum acceleration of M_1, expressed in units of g, is for this time interval the multiplier or design factor that is used for converting static stresses to maximum shock-induced stresses in infinitely stiff equipment components.

At the upper stops, there is another anvil-table velocity change similar to that produced by the hammer impact but this time in the negative or downward direction. The magnitude of this second velocity change is equal to the velocity at impact plus that velocity multiplied by a coefficient of restitution; hence it is dependent upon the initial anvil-table velocity change due to:

- Hammer impact,
- Anvil-table travel distance,
- Vibration phasing at the time of impact.

The shock effects of this second velocity change are then simply added to the shock motion already existing, and the principle of superposition permits evaluation of a second acceleration factor or design factor that is effective from the time the upper-stop impact occurs to the time when the anvil table again reaches its at-rest position. By this time the most severe parts of the shock disturbance have passed and any further anvil-table impact can be ignored.

The two design factors thus far obtained apply only to equipments that are infinitely stiff, whereas shock-induced stress in an equipment component with finite stiffness is dependent upon component natural frequency. A simple instrument—the reed gage—consisting of some cantilever-reed scribers mounted in a rigid frame, has been used for some time to provide frequency-sensitive "equivalent static accelerations" or "dynamic load factors." These are design factors applicable to systems having the same natural frequencies as the reeds and therefore serve as a basis for the design of actual equipment items. In a plot for a given shock motion, these acceleration factors versus reed frequency, constitute the familiar shock spectrum.

Now an analytical consideration of reed-gage response to the test-load motion is essentially a study of the response of a small mass-spring system attached to the load (Figure 2). Here m is mass, k is stiffness, and $\ddot{X}_R$ is reed acceleration. By means of Duhamels' Integral, derivations have been worked out for the acceleration response--or shock-spectrum response--of this reed mass to the shock-induced acceleration of test equipment M_1. For a specific test, the derived shock spectrum applicable after the hammer impact until the anvil-table reversal at the upper stops has a resonance-curve shape with its peak where reed frequency coincides with the natural frequency of the shock-machine setup.

After the upper-stop impact, spectrum height is dependent not only on frequency—as in the previous case—but also on the positions in their vibratory cycles of both the load mass and the reed mass at the time of impact. This vibration phasing is a very critical thing that has a powerful effect on spectrum height. It can produce a shock severity—after the upper-stop impact—that is increased substantially over what may

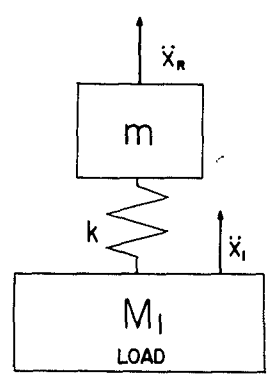

Figure 2 - Diagram on response of system to testing load

be caused by the main hammer impact. It can also cause a shock-severity reduction, but this is a possibility which is of no significance here.

By means of the two derived reed-response equations—one valid before and one after the upper-stop impact—points defining two curves of maximum reed acceleration versus reed natural frequency have been computed for each of three test-equipment weights—zero, 2,050, and 4,400 lb. The curves for the 2,500-lb load are shown in Figure 3. They indicate theoretical upper and lower bounds of shock-spectrum response when the machine is loaded to about one-half capacity. Here the upper curve—effective after the upper-stop impact—is approximately 50 percent higher than the lower or hammer-impact curve. This increase changes to about 100 percent for the very light (i.e., zero-weight) equipment and 25 percent for the 4,400 lb, whereas the lower curve is nearly the same for all three weights.

These derived curves—the pair shown here and those for the two other equipment weights—illustrate two instances of shock-test inconsistency:

First, each test-equipment component having a natural frequency the same as that of the shock machine setup is tested much more severely than components having nearby natural frequencies—either higher or lower.

And second the timing of the upper-stop impact can result in a shock severity that is increased to as much as twice that produced by the hammer impact.

The points spotted-in on Figure 3 are reed-gage spectrum values experimentally obtained during calibration tests with a load of 2,050 lb—the weight for which the two curves were computed. In the abscissa symbol ω/Ω, ω is reed natural frequency and Ω the natural frequency of the shock-machine setup. When this ratio is greater than 3 (this corresponds to a reed natural frequency of about 200 cps) the experimental points tend to be higher than the plotted boundary curves—apparently an effect of higher-mode disturbances neglected in the assumption of a simple two-mass-and-a-spring system. At the lower frequencies, however, agreement is fairly good. This, and a similar agreement of experimental points when the test load was 4,400 lb, signifies that for the lower frequencies the shock machine does behave like the assumed two-mass-and-a-spring model.

No attempt has been made here to evaluate the accuracy with which the machine simulates in the laboratory typical shock encountered in

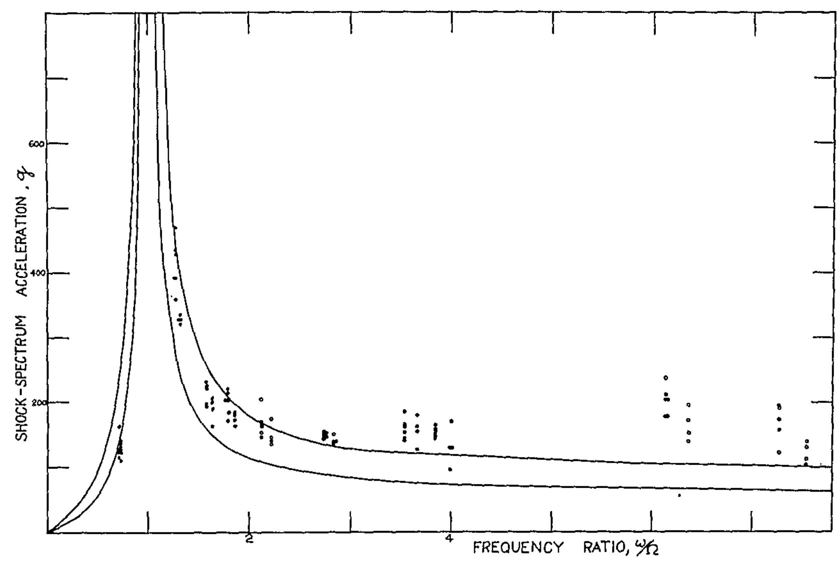

Figure 3 - Reed-gage spectrum values

the field. Some information on simulation accuracy, further operational characteristics, pertinent construction features, and presentation of additional reed-gage data will be available upon publication of the NRL report "Shock-Spectrum Characteristics of Navy Medium-Weight High-Impact Shock Machine."

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PART II

RANDOM, COMPLEX, AND SINE WAVE VIBRATION TESTING

INTRODUCTION TO RANDOM MOTION: THE NATURAL ENVIRONMENT AND ITS SIMULATION

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What is random motion and how is it generated? The properties of random motion most useful to the vibration engineer are stated in simple terms. The method used to create these motions in the laboratory and the operation of the major components of the required equipment are briefly described.

INTRODUCTION

Random accelerations, or a combination of random and periodic accelerations, have been found in many new environments, particularly in missiles and high-speed aircraft. The design of components which are to operate in such environments should consider the properties of random motion and should be verified by random-motion testing in the laboratory. Those components which survive an adequate random motion test in the laboratory can be expected to operate reliably in the actual environment.

This discussion presents the random motion concepts needed to design and test components to live in these random environments. The first part states the properties of random environments needed for design and test and describes the response of the component to the motion of the environment. The second part outlines the methods used to solve the problems encountered in random motion testing and describes the operation and characteristics of the more important items of the laboratory equipment.

Since this discussion is an introduction to random motion, it surveys the subject without delving deeply into any one phase. Only those properties which may be of use to a practicing engineer are presented, with little reference to the mathematical derivation of the results. Although many of the properties may be precisely stated in elegant mathematical forms, these forms have been avoided because they have little meaning to the average engineer. An alternate approach is taken, in which the important characteristics of random motion are described in terms of the responses of physical systems. This approach makes the engineer's first encounter with the subject of random motion more rewarding and in no way detracts from the generality of the results.

RANDOM MOTION ENVIRONMENT: THE EXCITATION

What is random motion? The most obvious characteristic of such motion is that it is non-periodic. For a periodic motion, such as a sinusoidal motion, the acceleration at a future instant is determined completely by the motion in the past. For a random motion this is not true. A knowledge of the past history is adequate to predict the relative probability of occurrence of various accelerations, but it is insufficient to predict the precise magnitude at some specific instant.

Random motions are not erratic in the common sense, but follow a very definite law.

These motions are the result of a very large number of events occurring by chance. In a missile or aircraft they are caused by the independent actions of the gas molecules in the exhaust and around the structure. The cumulative effect of such a very large number of independent actions is a motion that has specific characteristics and may be described by a normal probability law. The occurrence of certain types of motions is highly probable and the occurrence of others very improbable. As an example, there is little chance that three complete cycles of a sinusoidal motion will occur, but a very high probability that the motion will reverse its direction.

Although the magnitude characteristics of all random motions are governed by the same law and may be described in identical terms, the time or frequency characteristics of various random motions differ to a considerable degree. Graphs of the instantaneous acceleration as a function of time for arbitrary random motions can be made to have identical amplitude characteristics by a proper choice of the acceleration scale, however, no simple modification of the time scale will make the time characteristics similar.

A typical random motion is shown in Figure 1. On this graph the instantaneous acceleration of a typical random environment is plotted against time. Some acceleration peaks have a much greater magnitude than others and the period of time between values of zero acceleration varies irregularly. Most of the time the acceleration is at a relatively low level; only at rare intervals does the acceleration reach high values.

Although the random acceleration plotted in Figure 1 appears completely erratic, both amplitude and frequency characteristics of this motion can be stated. These characteristics can be described in terms of a property called acceleration density. If the acceleration density is known as a function of frequency, other properties of a random motion may be calculated. A plot of acceleration density vs frequency defines a steady-state random motion as completely as it can be known.

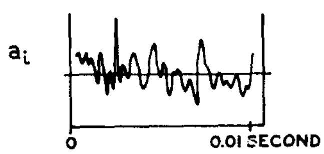

Figure 1 - Typical random motion instantaneous acceleration

The name acceleration density for this property of a random motion has not yet achieved general acceptance. Complete descriptive titles would be mean-square acceleration spectral density or mean-square acceleration per cycle. A plot of this property as a function of frequency would be called a mean-square acceleration density spectrum. Since these long names are cumbersome, various short names have appeared. The property has been called power density by analogy to the power which would be dissipated in a resistance if the acceleration were voltage across the resistor. Abridged forms of the long titles have been used, such as mean square density, spectral density, and acceleration density. The name acceleration density seems preferable because it is both descriptive and concise.

Most persons find the acceleration density concept somewhat difficult to grasp. To give meaning to this concept we may construct a hypothetical experiment to illustrate this property in terms of more familiar characteristics. The diagram of Figure 2 illustrates this hypothetical experiment.

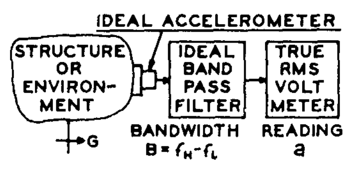

Figure 2 - Hypothetical experiment to illustrate acceleration density concept.

$$G = \underset{B \to 0}{\text{LIMIT}} \frac{a^2}{B} \quad (1)$$

The block to the left represents the environment. Since in most cases the motional environment of a component will be the motion of the structural of an airframe, the block is also labeled structure. This structure moves randomly with an acceleration density G.

Attached to the structure is an ideal accelerometer which produces a voltage proportional to the acceleration of the structure and applies it to an ideal band-pass filter.

The block in the center represents the band-pass filter. The output voltage of such a filter

reproduces the input voltage for all frequencies in the pass band between a low cutoff frequency f_L and a high cutoff frequency f_H. The frequency components of the input voltage which are lower than f_L or higher than f_H are not reproduced at the output. The frequency difference between f_L and f_H is the band width B of the filter.

The block to the right represents a true root-mean-square or rms voltmeter which reads the output voltage of the band-pass filter. The reading "a" of such a meter is proportional to the effective heating value or power producing value of the input voltage wave and is the normal value used for electrical power. A thermocouple type voltmeter is such an rms voltmeter.

We intend to give a physical meaning to the concept of acceleration density by relating it to the reading of the rms voltmeter and to the pass band of the filter. It is particularly instructive to consider what happens to the reading of the meter as the filter band width is varied.

Before we describe the operation of this system for random motion it is desirable to calibrate the system in terms of the more familiar sinusoidal units. Let us allow the structure to move with a sinusoidal acceleration rather than with the random acceleration density G shown on Figure 2. Let us maintain a sinusoidal acceleration of 1 g rms which is 1.41 g in the peak acceleration units generally used for sinusoidal accelerations. If the frequency of such motion is within the pass band of the band-pass filter, we can calibrate the rms voltmeter to read the value 1. By this procedure the voltmeter indication is made to read directly the rms value of the acceleration of the environment. As the frequency of the sinusoidal motion is varied with the amplitude fixed at 1 g rms, the meter will continue to read 1 so long as the frequency of the sinusoidal motion of the environment is within the pass band of the filter but will drop to zero whenever the frequency varies either below or above the pass band of the filter.

If we now allow the structure to move randomly we may use this hypothetical system to illustrate many characteristics of random motion. The accelerometer will produce a random voltage. The frequency components of this voltage within the pass band of the filter will be applied to the voltmeter. The indication "a" of the voltmeter will be the random acceleration of the structure in rms g for the frequency band of the band-pass filter.

If we decrease the band width of the band-pass filter, the voltmeter indication will decrease. This decrease indicates that the rms random acceleration for a narrow band of frequency is less than for a wide band of frequency. This fact, that the rms random acceleration is a function of band width, may be contrasted to sinusoidal motions in which the acceleration is independent of band width.

As the frequency pass band of the filter is reduced, the meter indication not only decreases, but decreases smoothly. This smooth decrease indicates that the acceleration of the environment is distributed over the frequency spectrum rather than concentrated at discrete frequencies. As the band width is narrowed, information from portions of the frequency spectrum is being prevented from reaching the meter.

As the band width is reduced to very narrow values the indication on the meter will start to wander about a mean value. The needle fluctuates more slowly as the band width of the filter is reduced. Although the fluctuation of the needle is not periodic, the rough frequency of fluctuation is approximately the band width of the filter in cps.

If the band width is reduced by successive factors of four, the rms random acceleration will drop by factors of approximately two. This relationship will be found to hold exactly as very narrow band widths are approached. This result indicates that, for narrow band widths, the square of an rms random acceleration is proportional to the band width.

We wish to give meaning to the property called acceleration density. This property may be defined in terms of the rms acceleration and of the band width. If we calculate the quantity a^2/B, which is the square of the rms acceleration divided by the band width, we will have a quantity approximately equal to the acceleration density. It was observed above that, for narrow band widths, the square of an rms random acceleration is proportional to the band width. Therefore, as the band width B is decreased, the quantity a^2/B will approach a constant value. That limiting value is the acceleration density G. This result is expressed by Equation (1):

$$G = \underset{B \to 0}{\text{LIMIT}} \frac{a^2}{B} \tag{1}$$

in units of g^2/cps.

Two plots of acceleration density as a function of frequency are shown on Figure 3. An

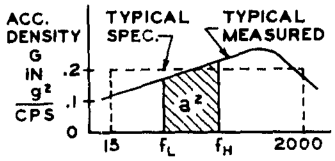

Figure 3 - Acceleration density spectra

$$G = \lim_{B \to 0} \frac{a^2}{B} \quad (1) \qquad a = \sqrt{\int_{f_L}^{f_H} G df} \quad (2)$$

$$a = \sqrt{BG_o} \quad (3)$$

acceleration density which might be measured in a typical random environment is shown by the solid line. In most environments, the acceleration density is not constant, but is a function of frequency. In any airframe both the level and the shape of the acceleration density spectrum vary from point to point within the structure.

The dashed line of Figure 3 is a typical test specification. The shape of this specification has been chosen because the environment it represents is simple to reproduce in the laboratory and is a logical compromise between the many different actual environments. An environment like that of this specification, which has an acceleration density constant as a function of frequency, is called a flat random motion spectrum or a white random motion spectrum. For convenience we shall designate a flat random motion spectrum by the symbol G_o. For such a flat spectrum the limiting process of Equation (1) is unnecessary; the acceleration density calculation is independent of band width.

Knowing the acceleration density as a function of frequency, the rms acceleration for any frequency band may be calculated. If the rms acceleration for the frequency band between f_L and f_H is desired, it may be calculated from Equation (2):

$$a = \sqrt{\int_{f_L}^{f_H} G df} \qquad (2)$$

In other words, the rms acceleration is the square root of the area under the curve of Figure 3 between the frequencies f_L and f_H. The square of the rms acceleration is illustrated by the crosshatched area in Figure 3. As the band width of the instrumentation is increased, the observed rms acceleration will continue to increase until the area under the acceleration density curve is completely included.

For the white random vibration case, in which the acceleration density is independent of frequency over this frequency band, Equation (2) for the rms random acceleration may be simplified to Equation (3):

$$a = \sqrt{BG_o} \qquad (3)$$

in units of rms g. This simple case, in which the acceleration density is independent of frequency, is the one shown by the dashed curve of Figure 3, and is the one commonly used for random vibration testing.

A sample calculation will illustrate the use of Equation 3. Consider a random motion test specification which calls for a constant acceleration density of 0.2 g^2/cps over the band width from 15 cps to 2,015 cps. The rms acceleration is then calculated in the following manner:

$$a = \sqrt{BG_o} = \sqrt{2,000 \times 0.2} = \sqrt{400} = 20 \text{ g rms} \qquad (4)$$

The curve of Figure 4 is identical to the curve of Figure 1. Now that the term rms acceleration has meaning, the vertical axis can be marked off in units of the rms acceleration as calculated from Equation (2). Although the precise acceleration which will occur at any future instant cannot be calculated, the probability of that acceleration occurring can be stated. Most of the time, the instantaneous acceleration will be low, in fact it will be between the rms values of plus one and minus one 68 percent of the time. The curve of Figure 4 stays between these values about 68 percent of the time. The magnitude of the instantaneous acceleration will exceed twice the rms value about 5 percent of the time and 3 times the rms value about 0.3 percent of the time. These percentages are shown by the arrows to the right of the curve.

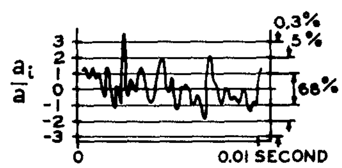

Figure 4 - Typical random motion instantaneous acceleration divided by rms acceleration

The curve of Figure 4 represents a wide band random acceleration. Such a curve could represent the example calculated above for a constant acceleration density of 0.2 g^2/cps between 15 and 2015 cps. If this curve were to represent that example, the rms value would be 20 g and one peak is shown which exceeds 60 g.

In a narrow band random motion the probability relationship between the instantaneous acceleration and the rms acceleration is the same as for a wide band motion. However, if the band width is sufficiently narrow, for example less than 20 percent of the center frequency of the band, the appearance of a plot of the instantaneous acceleration as a function of time will differ considerably from the curve of Figures 1 and 4. Although such a narrow band motion will still be random, it will take on many of the characteristics of a sinusoidal motion. The plot will be approximately sinusoidal, with a general frequency about the same as the center frequency of the narrow band, but will have a constantly varying amplitude and phase. The curve of Figure 6 is a narrow band random motion.

The preceding paragraphs have discussed a fundamental property of random motion called acceleration density. This property is particularly important in the description of a random motion environment. If the motion of an environment can be described by stating the acceleration density as a function of frequency, Equations (2) and (3) permit the calculation of the rms random acceleration for any desired frequency band.

We have so far been discussing the random motion of an environment. Since in a typical case the environment might be the structure of a missile, the motion of the environment may be considered to be the motional excitation applied to a component or specimen attached to that structure.

In the section to follow, we shall turn our attention from the motion of the environment to the motion or response of a specimen attached to that environment.

RANDOM MOTION ENVIRONMENT: THE RESPONSE

What happens to a specimen subjected to random motion? It is informative to investigate the response of a simple system to a random motion excitation. The response of a resonating specimen is particularly useful.

The system sketched in Figure 5 represents either a single degree of freedom system or a normal mode of a more complex object. The mass, spring, and damper shown represent the equivalent mass, spring force, and damping force for this mode of vibration. The structure to the left represents the environment. The motion of this structure is the excitation for the resonating system.

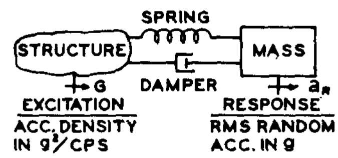

Figure 5 - Specimen response to random excitation

$$a_R = \sqrt{\frac{\pi}{2} G f_o Q}$$

To illustrate our terminology, it is useful to consider the response of this system to an excitation which is sinusoidal rather than random. If the frequency of the sinusoidal excitation is low, the spring will be essentially a rigid connector and the mass will move with the excitation. At a very high frequency, the inertia of the mass will be great and the motion of the mass will be small. At an intermediate resonant frequency f_o, the motion of the mass may build up to a magnitude considerably greater than that of the excitation. The maximum ratio of the motion of the mass to the motion of the structure for sinusoidal excitations is called the peak magnification factor, or Q, of the resonance. The peak magnification factor Q is also equal to the reciprocal of twice the ratio of the actual damping to the critical damping for this normal mode.

If the motion of the structure is random rather than sinusoidal, the system responds differently. The motion of the mass will be random and will have the characteristics of narrow band random motion. If the excitation has an acceleration density G in the region of the resonant frequency, the rms acceleration response of the mass is given by Equation (5):

$$a_R = \sqrt{\frac{\pi}{2} G f_o Q} \qquad (5)$$

where a_R is the rms random acceleration response in g, G is the acceleration density in units of g^2/cps, f_o is the resonant frequency of

the normal mode in cps, and Q is the peak magnification factor of the normal mode for sinusoidal excitations.

It may be noted that the rms random acceleration response is a function of the resonant frequency, which is not true for sinusoidal excitations. It may also be noted that random response is proportional to the square root of Q rather than to Q as is the case for sinusoidal excitations.

The instantaneous acceleration response of a resonant specimen is shown in Figure 6. Since the rms acceleration response can be calculated by Equation (5), the vertical scale has been generalized by dividing the instantaneous acceleration by the rms acceleration. The magnitude of the instantaneous response as measured on the vertical scale then gives the acceleration of the mass as a fraction of the rms acceleration. The curve shown is for a typical structure resonant at 150 cps with a Q near 8. It may be noted that the waveform looks much like a 150 cps sine wave with a constantly varying amplitude and phase.

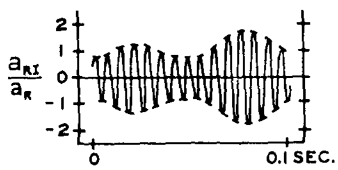

Figure 6 - Instantaneous acceleration response of a randomly excited resonant specimen

The instantaneous acceleration of the mass for this resonant response to a random excitation is very different from the wide band random acceleration drawn in Figure 4. Although the two curves do not appear similar, the probability of occurrence of specific accelerations for the two curves is the same. As in Figure 4, the acceleration response is less than the rms value 68 percent of the time and exceeds three times the rms value only 0.3 percent of the time. The response of a single degree of freedom system, which has very little damping, is similar to the acceleration of an environment viewed through a narrow band-pass filter.

A dashed curve has been drawn on Figure 6 connecting the peaks of the instantaneous acceleration responses with a simple curve called an envelope. The magnitude of the envelope fluctuates in a somewhat erratic manner but an approximate fluctuation frequency may be observed. This fluctuation frequency is approximately equal to the resonant frequency divided by the peak magnification factor or Q of the normal mode in question. This fact may be useful in estimating the Q of a resonance.

The preceding sections have discussed the physical motion of objects. The random motion of an environment such as the structure of a high-speed aircraft or missile was described first, followed by a description of the response of a specimen to the motion of that environment.

The simulation of this motional environment in the laboratory will be discussed in the sections to follow. If the motion can be simulated, the performance and endurance of a specimen can be evaluated during design. Specifications for vibration tests can then be established which will insure that the specimen will give satisfactory performance in its normal environment. The general method used for such simulation will be outlined first, followed by a discussion of the operation of the major components of the equipment used to perform this simulation.

SIMULATION OF RANDOM ENVIRONMENT: THE METHOD

The objective of laboratory simulation is to create a motion similar to that found in actual observed environments. This motion can then be used for design evaluation or qualification testing. To create this motion, we first decide what acceleration the specimen should have. We then obtain a voltage proportional to that acceleration. Finally, we adjust the laboratory equipment to create an acceleration proportional to that voltage.

The laboratory equipment required to perform this simulation is called a Complex Motion System and consists of three major components. A vibration exciter shakes the specimen, a large electronic amplifier powers the vibration exciter, and a Complex Motion Console supplies a modified voltage to the amplifier to create a vibration table acceleration proportional to an input voltage. Sources of input voltage proportional to commonly desired accelerations are mounted in the console and others may be externally connected as desired.

A specimen and the three components of such a system are shown in Figure 7. The specimen

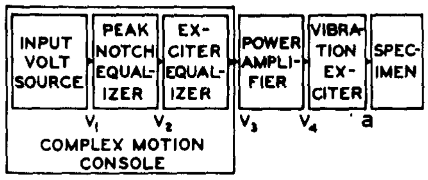

Figure 7 - Major components of complex motion system

being tested is at the far right. It is being vibrated by a vibration exciter which in turn is being driven by a power amplifier. The amplifier receives a modified voltage from the console which contains among other things, the input voltage source, the Peak Notch Equalizer and the Exciter Equalizer shown. In the block diagram of Figure 7, the letter "a" represents the acceleration of the specimen. The voltage from the input voltage source is called V_1. The voltages at other major points in the system are labeled on the block diagram. It is desired that the acceleration of the specimen exactly reproduce the input voltage V_1.

We can define the gain of a vibration exciter as the ratio of the acceleration of the table and specimen to that voltage which drives the vibration exciter. Unfortunately, the gain of the vibration exciter varies very greatly with frequency. This fact severely complicates the problem of creating an acceleration proportional to the input voltage.

If an acceleration proportional to the input voltage is to be obtained, the gain of the entire system from the input voltage V_1 to the specimen acceleration "a" must be a constant and not a function of frequency. The equalizers in the Complex Motion Console are supplied to make this gain constant. When the gain of these equalizers is adjusted to be the inverse of the gain of the vibration exciter, the gain of the entire system becomes constant and the acceleration reproduces the input voltage.

The use of two types of equalizers may be explained by referring to the two reasons why the gain of the vibration exciter varies. First, the gain of the vibration exciter is not basically constant to start with, even without a table load, and secondly, each resonance of the specimen introduces a sudden variation in gain. The Exciter Equalizer is provided to compensate for the gain variation of the basic vibration exciter as modified by the non-resonant portion of the load. It may be adjusted to have "a" gain the inverse of the vibration exciter gain.

A Peak Notch Equalizer is provided to compensate for the gain variation due to a specimen resonance. Each of the Peak Notch Equalizers has a gain characteristic as a function of frequency which is the inverse of the variation in gain of the shaker due to a resonance in the specimen. A number of Peak Notch Equalizers are normally used, one for each of the major resonances of the specimen. When the Exciter Equalizer and the Peak Notch Equalizers are properly adjusted, the gain of the Complex Motion Console from V_1 to V_3 is the inverse of the gain of the vibration exciter from V_4 to the table acceleration "a". Since the amplifiers used for Complex Motion System operation have a very flat frequency response, the ratio of V_4 to V_3 is a constant. Since the amplifier gain is constant and the gain of the equalizers compensates for the gain of the vibration exciter, the over-all system gain is constant and the acceleration is proportional to the input voltage.

As soon as the system gain is constant it is possible to begin random motion testing. We may use a random voltage generator for the input voltage source, and since the acceleration is proportional to the input voltage, the specimen will be subjected to random accelerations.

This article discusses only those components of a Complex Motion System needed for an understanding of the basic operation. These components are shown on Figure 7. Other components are usually added to the system to increase its convenience and flexibility and to protect the system and the specimen from component failure and operator abuse. Such other components include an acceleration limiter which prevents high acceleration peaks from reaching the specimen, a displacement limiter which prevents the vibration exciter from hitting the mechanical stops and jolting the specimen, a variable band-pass filter which adjusts the band width of the input voltage, an rms voltmeter which reads the rms magnitude of any voltage or acceleration, accelerometers and cathode followers used to convert motion to voltage, an oscilloscope to view waveforms, equalizer adjustment instrumentation, and an amplitude protector which prevents system failures from damaging the vibration exciter or the specimen.

SIMULATION OF RANDOM ENVIRONMENTS: OPERATION OF EQUIPMENT

We have so far discussed the general operation of a Complex Motion System. It is now

advisable to look at each one of the major components in somewhat greater detail.

If random motion testing is contemplated, a vibration exciter should be chosen which is suitable for the intended application. In particular, it should have a displacement range adequate to handle the range of expected displacements and it should have a force rating great enough to produce the required rms and peak accelerations for the range of test loads expected. Unfortunately, neither the peak-to-peak displacement nor the peak acceleration of a random motion is a well defined quantity. The displacements are random if the accelerations are random and peak displacements exceed three times the rms value of displacement the same fraction of the time as peak accelerations exceed three times the rms value of acceleration. This fraction is 0.3 percent. It has become standard practice to limit both displacements and accelerations to three times the rms value. This means that for 0.3 percent of the time the accelerations may be limited and for another 0.3 percent of the time the displacements may be limited. The choice of three times the rms value for a limiting level is a completely arbitrary one. However, it is necessary to make some such arbitrary choice if the force and displacement range of the equipment are to be practically realizable.

The displacement range of a specific vibration exciter will be adequate for a vibration test if that range exceeds the peak-to-peak displacement of that test by a satisfactory margin. If the displacement is limited to three times the rms value in each direction, the peak-to-peak displacement is equal to 6 times the rms value of displacement. The peak-to-peak displacement defined in this manner may be calculated from the acceleration density. For a wide-band, flat acceleration density spectrum the peak-to-peak displacement is given by Equation (7):

$$d_{pp} = \sqrt{\frac{1,160\ G_o}{f_L^3}} \tag{7}$$

where d_{pp} is the peak-to-peak displacement in inches, G_o is the acceleration density in units of g^2/cps, and f_L is the lower cutoff frequency of the applied acceleration density spectrum. If the acceleration density spectrum is not flat, Equation (7) may be used to calculate an approximate value for the peak-to-peak displacement by replacing G_o by the value of the acceleration density G in the region of f_L.

The rms random force rating of a vibration exciter should exceed the rms random force requirement of the vibration test. This rms random force requirement is a function of the rms random acceleration desired and of the mass of the object being tested. The rms random force for a vibration exciter is defined in a manner analogous to the definition of a peak sinusoidal force as the product of the total moving weight and the rms random acceleration. Equation (8) states this relationship:

$$F = Wa \tag{8}$$

where

F is the force in rms random pounds,

W is the weight in pounds of the entire moving system including both specimen and vibration exciter moving element assembly,

a is the acceleration of the moving system in rms random g.

The rms force rating of a vibration exciter is determined primarily by heating effects within the vibration exciter. This rating is the same as the rms sinusoidal force rating and is equal to 0.707 times the peak sinusoidal force rating. For random motions, peak accelerations equal to at least three times the rms acceleration are desired. For this reason, peak forces three times the rms force are also necessary. Vibration exciters for random motion use are designed to produce peak forces equal to three times the rms force rating.

The choice of an amplifier for a random motion application is determined by the random force to be produced and the characteristics of the particular vibration exciter used to produce this force. Although adequate power is the primary consideration, a flat frequency response and low distortion are also important factors. The amplifier must be carefully selected since it is usually the most expensive single item in the system.

For random motion testing, the amplifier must be designed for the waveshapes actually encountered in use. In particular, the amplifier must be capable of producing the high voltage and current peaks required to produce forces equal to three times the rms force rating of the vibration exciter. Designs based on sinusoidal waveshapes are either unsuitable for random motion testing or result in amplifier sizes much larger than those required for amplifiers designed specifically for the random motion application.

The frequency response of the amplifier must be flat if the operation of the equalizers

is not to be affected, and should be accompanied by a good transient response to reproduce the waveshapes encountered in operation.

The amplifier must have low distortion to prevent the undesired excitation of any resonance by distortion products. These distortion products are most serious when high signal levels are required to produce frequency components which are submultiples of a frequency at which the vibration exciter gain is high. Unfortunately, this serious condition is present at submultiples of every severe specimen resonance.

The two systems chosen for the various tests are probably the smallest and largest sizes currently popular. (Systems smaller, between, and larger than these also are available.) The smaller system uses a vibration exciter with an rms random force rating of 840 lb, an amplifier with a 10 KVA capacity, and will drive a specimen weight or payload of 25 lb to an rms acceleration of 20 g. This is the rms level used in the example calculated above which had an acceleration density of 0.2 g^2/cps and a band width of 2,000 cps. The larger system listed, which has a 5,000 lb rms random force rating, requires a 60 KVA amplifier and drives a 155 lb table load to the same rms acceleration. Each of the systems shown will produce peak forces up to three times the rms force listed. The forces shown are calculated for a band width of 2,000 cps as was used in the example for which the 20 g rms acceleration was calculated. These forces are approximately correct for narrower band widths as well.

If the force rating of a system is inadequate to drive the specimen to the desired acceleration density over the entire 2,000 cps band, it is possible to divide up the band into two or more frequency ranges and test each range separately. By this procedure, it is possible to achieve acceleration densities in a restricted frequency range considerably greater than would be possible if the entire frequency band were tested at the same time. For example, if the larger of the two systems shown were available, and if it were desired to test a 300 lb object to an acceleration density of 0.2 g^2/cps, the system would be too small to test the entire frequency band from 15 to 2,015 cps at one time. If, however, this test were broken into four tests, each for a 500 cps wide band, the acceleration density of 0.2 g^2/cps could be obtained. This procedure is based on the assumption that a test which successively tests four separate 500 cps frequency bands is a valid test. Since such an assumption cannot usually be verified, the use of this procedure can only be justified in an emergency.

In the Complex Motion Console are mounted most of the components which make random motion operation possible. Among these components are the Peak-Notch Equalizers and the Exciter Equalizer which compensate for the gain variations of the vibration exciter—specimen combination. A simplified block diagram of the entire complex wave system is repeated above Figure 8(a). The curves show the gain-frequency characteristic of the vibration exciter. This is the characteristic previously discussed and for which the equalizers must compensate.

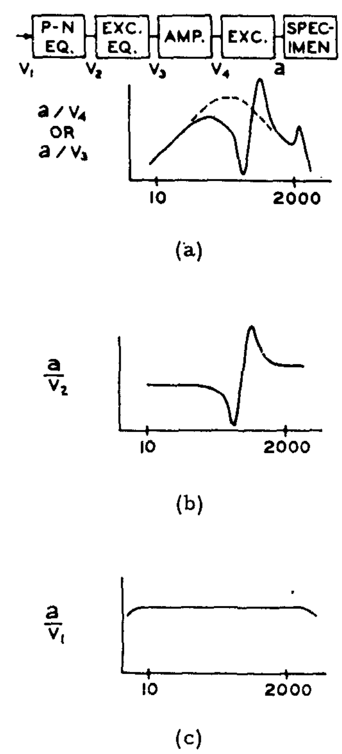

Figure 8 - Frequency response equalization; (a) unmodified exciter response, (b) with exciter equalizer, (c) with both equalizers

The dashed curve of Figure 8(a) represents the gain characteristic of the unloaded vibration exciter. Although the gain characteristics in the low frequency range are not constant, the variation with frequency is relatively gradual.

You will note, however, that a peak occurs at a high frequency. This peak occurs for all vibration exciters and is due to an axial mode resonance within the moving structure itself in which the center of the table moves in opposition to the motion of the main driving coil. As the frequency is raised above this resonant frequency, it becomes increasingly difficult to supply an adequate force to the specimen. A practical upper frequency limit to operation is about twice this resonant frequency and, whenever possible, the frequency band of operation should be kept below this frequency.

In Figure 8(a) the curve drawn with a solid line shows the gain of a vibration exciter on which is mounted a specimen with a single severe resonance. The resonance of the specimen introduces a large notch and peak to the gain characteristic. At high frequencies the resonating mass of the specimen is effectively isolated and the gain curve approaches the gain for the unloaded vibration exciter. At low frequencies, the specimen looks like a heavy dead load applied to the table and the same acceleration requires larger voltages than for the unloaded case. As the frequency is increased through the resonant frequency of the specimen, the gain first drops to a minimum and then rises to a peak. The gain minimum occurs at the resonant frequency of the specimen. Since the specimen acts like a dynamic vibration absorber at this frequency, large forces and large voltages are required to obtain the same table acceleration as at lower frequencies. The gain maximum occurs at the frequency of a combined resonance of the specimen and moving assembly, and very little force is required to obtain large accelerations.

The Exciter Equalizer shown in the block just to the left of the amplifier compensates for the characteristics of the vibration exciter by itself, independent of the effect of the resonating load. In other words, the Exciter Equalizer gain is the inverse of the gain shown by the dashed curve in Figure 8(a). The Exciter Equalizer compensates both for the gradual slopes in the low-frequency region and for the sharp peak in the high-frequency region. This is accomplished by creating, in the Exciter Equalizer, the exact reciprocal of the transfer function of the unloaded vibration exciter and attached non-resonant mass.

The curve of Figure 8(b) shows the gain from the input of the Exciter Equalizer to the acceleration of the specimen. The change from the solid line of Figure 8(a) to the curve of Figure 8(b) illustrates the compensation due to the Exciter Equalizer. Although the curve on Figure 8(b) still has the notch and the peak caused by the resonance of the specimen, the general shape of the vibration exciter gain curve is no longer present.

The Peak-Notch Equalizer shown just to the left of the Exciter Equalizer completely compensates for the fluctuation in the vibration exciter gain-frequency curve caused by a single-specimen resonance. This means that the Peak-Notch Equalizer gain-frequency characteristic may be adjusted to be inverse of the gain variation caused by a specimen resonance such as the one illustrated in Figure 8(b). The Peak-Notch Equalizer compensates for both amplitude and phase variations, not only in the vicinity of resonance, but also over the full frequency band of operation. This complete compensation is achieved by creating, in the Peak-Notch Equalizer, the exact reciprocal of the modification to the vibration-exciter transfer function caused by a resonance in the specimen.

The curve of Figure 8(c) shows the gain from the input of the Peak-Notch Equalizer to the acceleration of the specimen. The change from the curve of 8(b) to the curve of 8(c) shows the compensation due to the Peak-Notch Equalizer. When properly adjusted, the curve is flat over the operating frequency range. Since most specimens will have a number of resonances, not just the one illustrated in Figures 8(a) and 8(b), a number of Peak-Notch Equalizers are used, all connected in tandem so that their gain-frequency characteristics multiply.

A proper adjustment of the equalizers has made the over-all system gain, from the input voltage to the specimen acceleration, independent of frequency over the operating frequency range. The table acceleration will now reproduce any voltage applied to the input. If a random acceleration is desired, the input voltage may be obtained from a random voltage generator. If the reproduction of the motion of an actual structure is desired, the input voltage may be obtained from a tape recording of the acceleration of that structure.

SUMMARY

Reliability is important. If a device is to perform its intended function, each of the components within that device must be reliable; and the more components there are, the more reliable they must be. Since vibration is a

major cause of failure, a rigorous vibration test program will significantly improve reliability.

Certain steps must be taken to establish a rigorous vibration test program for components which are to operate in random motion environments. Random motion studies of the structures on which these components are to operate will produce a set of curves, each somewhat similar to the solid line of Figure 3. Usually, the curves taken from various locations in the structure, from different structures, and at different times, will not be the same. A component test specification derived from these acceleration density curves is frequently made more severe than the expected environment of the component.

Before component testing for design evaluation or qualification testing can begin, a suitable Complex Motion System must be assembled. A review of the test specification and the weight of the component, the test fixture, and the vibration exciter moving element assembly will permit the calculation of the rms random force level and will suggest the choice of a vibration exciter, an amplifier, and a Complex Motion Console. As soon as the component and fixture are mounted on the vibration exciter, the equalizers may be adjusted to flatten the frequency response. If a random voltage generator is connected to the input, the component acceleration density may be raised to the test specification level. Random-motion testing may then continue for the time chosen for the test specification, or until the component prematurely fails.

A component which survives such a test should operate reliably in the actual environment. If all the components in a device survive similar vibration tests, and do not fail from other causes, the device should be reliable and can be expected to perform its intended function.

APPENDIX

SUMMARY OF EQUATIONS

$$G = \lim_{B \to 0} \frac{a^2}{B}; \qquad a_R = \sqrt{\frac{\pi}{2} G f_0 Q}$$

$$a = \sqrt{\int_{f_L}^{f_H} G\,df} \qquad d_{PP} = \sqrt{\frac{1160 G_0}{f_L^3}}$$

$$a = \sqrt{B G_0} \qquad F = W a$$

A1 - Random motion equation

IF

ACCELERATION DENSITY CONSTANT AT $G = 0.2\ g^2/\mathrm{CPS}$

LOW FREQUENCY $f_L = 15$ CPS

HIGH FREQUENCY $f_H = 2015$ CPS

THEN

$$B = f_H - f_L = 2015 - 15 = 2000$$

$$a = \sqrt{B G_0} = \sqrt{2000 \times 0.2}$$

$$= \sqrt{400} = 20\ g\ \mathrm{RMS}$$

A2 - Sample calculation of acceleration

PEAK TO PEAK DISPLACEMENT TO ± 3 TIMES RMS VALUE IN INCHES $\quad d_{PP} = \sqrt{\frac{1160 G_0}{f_L^3}}$

RANDOM FORCE IN RMS POUNDS $\quad F = W a$

W IN POUNDS; a IN RMS g

A3 - Vibration exciter requirements

* * *

EXPERIMENTS IN RANDOM VIBRATION

V. C. McIntosh and Neal Granick, WADC

The new concept of random vibration testing must be much better understood and its value determined before millions of dollars are expended for new equipment. To become better acquainted with the problems, a study of the response of simple beams to this form of vibration was undertaken and is described here. It is intended to continue and extend this investigation to determine the value of random vibration testing.

Upon superficial examination of the advantages being cited for random vibration, this new concept of testing takes on many attractive features. However, upon closer examination, the basic problems encountered during practical use of random vibration are at least as many and varied as those for sine wave vibration. To become better acquainted with some of these problems, a study of the responses of simple beams to this form of vibration was initiated. The motivating idea was that these basic problems might be best clarified by examining the response of a simple mechanical system to random vibration. It is intended that this work will be extended so as eventually to provide the answer to the following question facing our armed forces: "Is it necessary to adopt random vibration testing as a general requirement?" If the answer is "Yes," it means the expenditure of millions of dollars for new equipment and the obsolescence of much existing equipment.

BEAM FATIGUE TESTS

To illustrate one problem with random vibration, simple beams were tested to failure first with sine-wave testing and then with random testing. Considering the time required to produce failure by both methods, the sine-wave input was compared with the random input.

A series of SAE 8630 steel beams 1/32-in thick, 12-1/2 in long, and 1/2-in wide were clamped individually at the center to an MB Model C-31-1 electrodynamic vibrator. The beams had been carefully selected from a much larger group according to their surface perfection. A 6-in cantilever overhang extended on each side of the clamp. Sine waves were generated from an oscillator and fed to an amplifier which controlled the vibrator armature current. White noise random vibration was supplied by a General Radio Model 1390-A noise generator used in conjunction with a Krohn-Hite Model 330A filter and a Ballantine voltmeter used as a preamplifier to obtain sufficient vibratory excitation. The acceleration was obtained by differentiating the signal generator output with an MB Model M-3 vibration meter. Figure 1 shows a typical setup with the beam mounted on the vibrator and the auxiliary equipment used.

Several beams were vibrated with sine-wave excitation at the fundamental cantilever bending mode. The tip amplitude of the beams was held constant until a fatigue failure occurred. The time to failure was noted with the corresponding rms acceleration producing the failure. Similarly, several beams were subjected to random vibration at a fixed bandwidth and rms acceleration. The average times required to produce failures under both sinusoidal and random vibration are presented in Table 1 along with

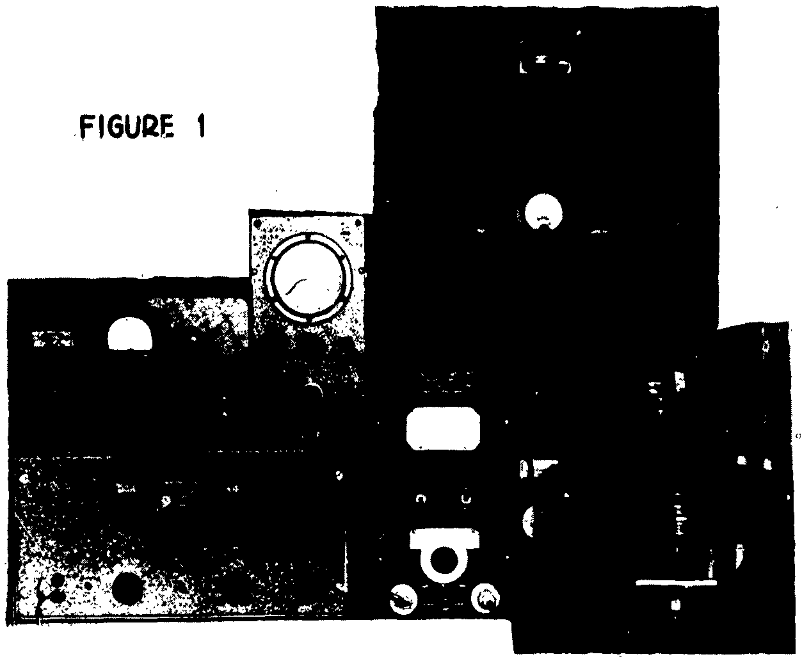

Figure 1 - General view of experimental arrangement showing beam mounted on vibrator and auxiliary equipment

TABLE 1
Comparison of Sine Wave With Random Vibration Results

Type of Vibration	No. of Samples Tested	Input (rms g)	Beam Tip Double Ampere (in).	Band Width (cps)	Average Time to Failure (min)
Sine Wave - 1st Mode	12	3.7	2.5		43
Sine Wave - 1st Mode	5	2.5	2.3		68
Sine Wave - 2nd Mode	3	6.3	0.25		66
Random	7	20.5		20 - 500	53
Random	3	15.1		20 - 200	48
Random	3	10.6		20 - 100	46

other pertinent information. Due to the limited time available, the number of samples was restricted to the minimum required to demonstrate consistent results. Another group of beams was subjected to sinusoidal vibration at the second bending mode to determine what average rms acceleration would produce comparable failure times.

BEAM FATIGUE RESULTS

There is, of course, a difference in the way fatigue damage is accumulated with sine-wave tests when compared with random vibration tests. For example, Figure 2(a) shows a beam subjected to random vibration and responding

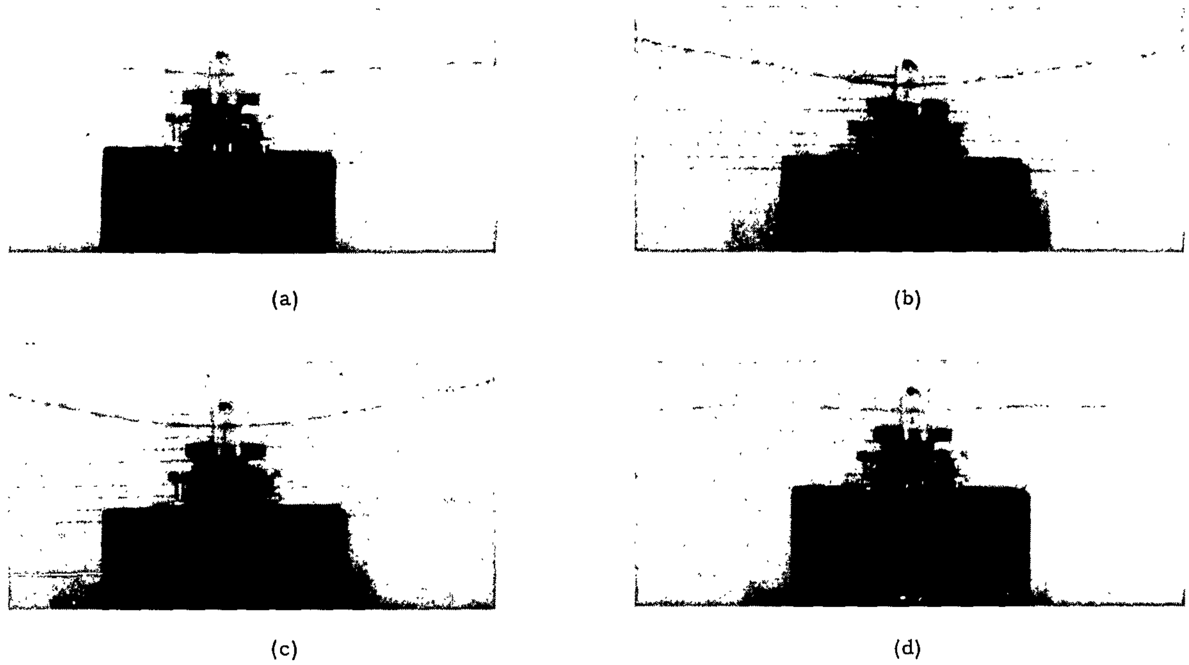

Figure 2 - Examples of beam response to sine-wave and random vibration excitation: (a) 20.5 g rms random vibration, 20 - 500 cps band width; (b) two-frequency vibration input, 27 cps and ±3.7 g, 170 cps and ±6.3 g; (c) ±3.7 g, sine-wave vibration input, first mode, 27 cps; (d) ±6.39, sine-wave vibration input, second mode, 170 cps.

with a combination first and second mode. Figure 2(b) shows a complex two-frequency sine-wave test on which first and second mode vibrations were superimposed and from which a similarity to the random response in Figure 2(a) is apparent. Figures 2(c) and 2(d) show the pure sine-wave response of the fundamental and second modes, respectively. Obviously, the damage from random vibration testing reflects a composite picture of the stresses resulting from superimposed higher order modes in addition to the fundamental mode. Nevertheless, the end result was an identical failure at the beam root produced alternately with random or sine-wave testing. Hence, a measure of the beam's ability to withstand repeated stressing was obtained equally as well by sine-wave vibration in either the first or second mode as was obtained with random vibration.

From Table 1 (sine wave - first mode), it may be interpolated that the acceleration required to produce a failure in 50 min is approximately 3 g rms. To obtain the same failure time with random vibration, an rms input acceleration of approximately 20 g was required in a band width of 20 to 500 cps. Hence, nearly seven times the acceleration and seven times the vibrator capacity were needed with random as compared to sine-wave vibration.

Effect of Band Width on rms Acceleration:

Another limitation illustrated by the beam-failure tests was that sufficient excitation to produce failure could not be obtained in band widths beyond 20 to 500 cps. If it were found necessary to increase the band width, for example, to an upper limit of 2,000 cps so as to include a known environment, the acceleration input ratio for this beam would be even greater. Figure 3 illustrates the high values of rms acceleration imposed by broad-band random vibration. The calculated rms acceleration curve is determined by the equation:

$$E = e_f \sqrt{BW} \qquad (1)^*$$

where

E = total rms acceleration in g units.
e_f = spectral density in g per $\sqrt{cps}$.
BW = band width through which white noise is applied.

*Kaufman, Joseph, "A Re-Evaluation of Vibration Testing Techniques," Electrical Manufacturing, November 1955

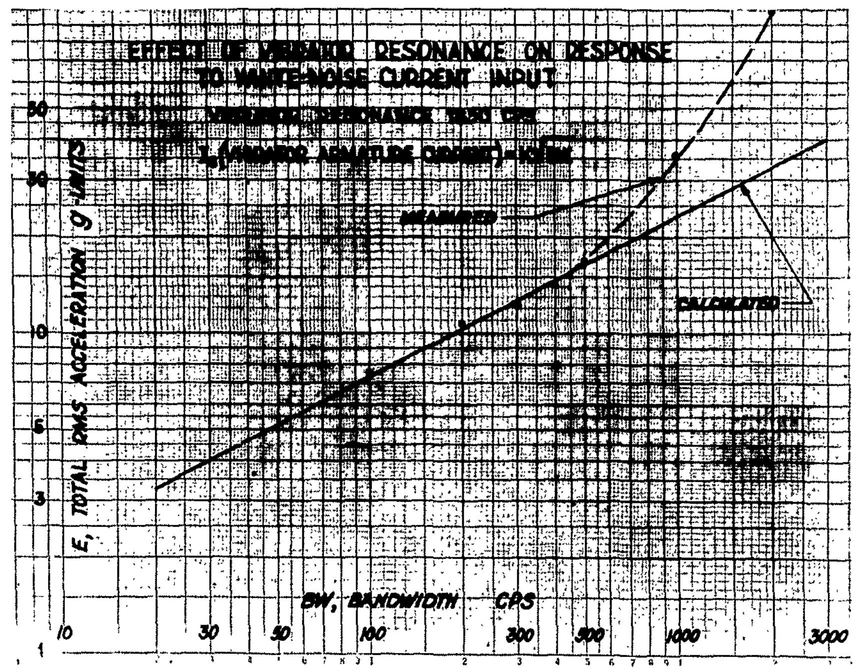

Figure 3

Usually a constant spectral density is specified which results in the calculated straight line relationship shown in Figure 3. This shows that with a constant spectral density an increase of the band width from 100 to 2,000 cps increases the rms acceleration requirement by a factor of 4.5 $\sqrt{2{,}000/100}$. In failing, the simple beams required a spectral density of approximately 1.1 g per $\sqrt{\text{cps}}$. These beams were more susceptible to failure than most items of equipment which often are required to withstand extended periods of ±10 g sine-wave testing. For example, spectral densities as great as 2 g per $\sqrt{\text{cps}}$ might be required to produce damage similar to that caused by the ±10 g sine-wave resonance tests. In a band width of 2,000 cps, a spectral density of 2 gives an over-all rms acceleration of 2 $\sqrt{2{,}000}$ = 90 g. This high level of acceleration is considerably beyond the capabilities of present vibration machines and hence imposes a severe limitation on the practical use of broad-band white noise for vibration testing.

VIBRATOR PERFORMANCE WITH RESONANT LOADS

Figures 4-7 illustrate in varying degrees the effects of resonant loads on vibrator performance. Beams weighing from 3 to approximately 1/2 times (1.8 to 0.25 lb) as much as the vibrator armature, which weighs 0.6 lb, were vibrated with a white noise of 20 to 1,000 cps applied to the vibrator. The acceleration output of the vibrator was fed into a Davies Model 510 spectrum analyzer (with an analyzer band width of 25 cps) to obtain the traces shown. Spectrum analysis was conducted throughout the frequency range of 0 to 2,000 cps. The gains were maintained constant during the recording of the three curves in each figure.

Figure 4 shows the extreme effects which can be produced by a resonant test item. Trace C shows the vibrator response resulting from white-noise excitation of a beam weighing 1.8 lb. Curve B was obtained by rigidly securing 1.8 lb of lead to the table. Curve A was obtained from the acceleration output of the shaker with bare table.

The high peaks in the vicinity of 100 and 700 cps occur at resonant frequencies of the beam. The peaks at 1,200, 1,400, and 1,850 cps result from shaker armature resonances. Note that the 1.8 lb rigid mass load reduced the shaker resonance from 1,850 to 1,200 cps. This illustrates the fact that a shaker system resonance may be above the test frequencies normally, but a heavy load may bring peaks within the testing range.

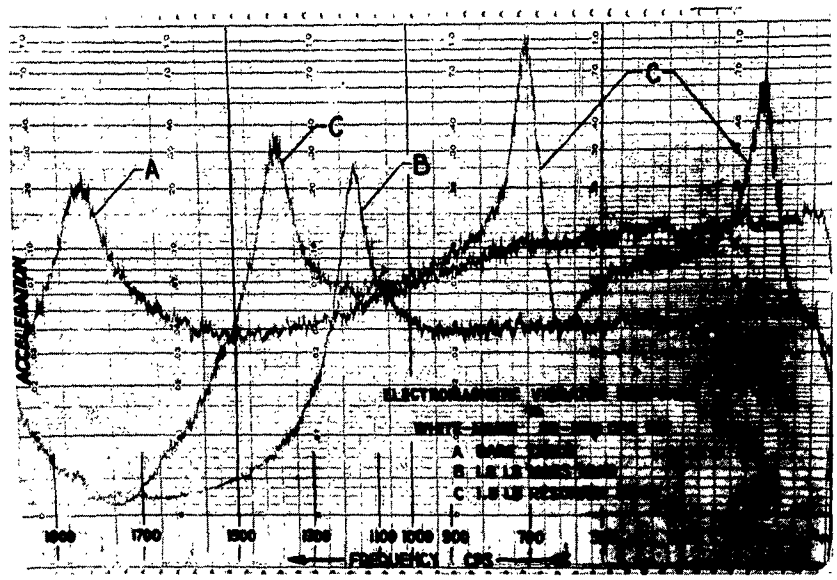

Figure 4

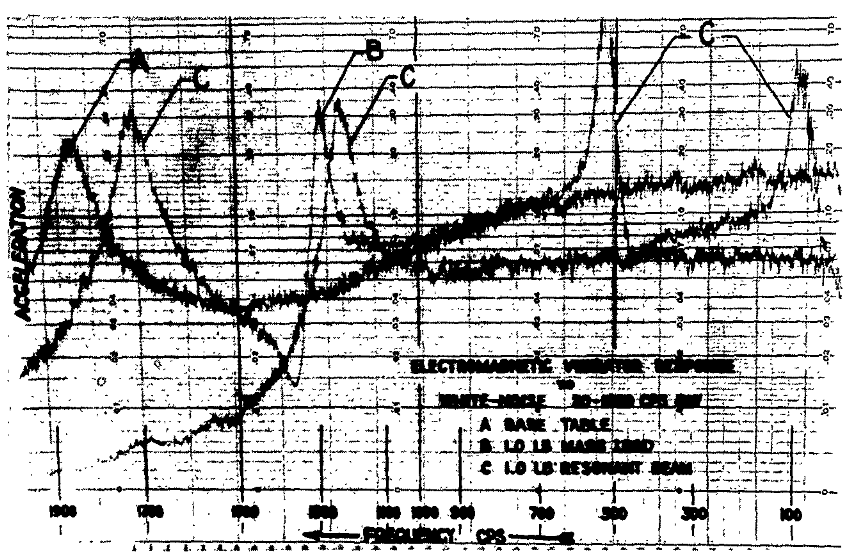

Figure 5

Variations in mass loading which are presented in Figures 4 through 7 show that the smaller resonant beams have correspondingly less effect on the shaker response. However, the smallest of the test beams (0.25 lb) where the ratio of the beam weight to the armature weight was less than 1/2 resulted in a peak rms response of 7 times the value resulting from a rigid load. Often the test specimen weight exceeds that of the vibrator armature and specimen resonances may result in peaks comparable to those shown in Figures 4-7.

The apparent solution to this problem is to introduce band-pass and rejection filters to compensate for the response characteristics of

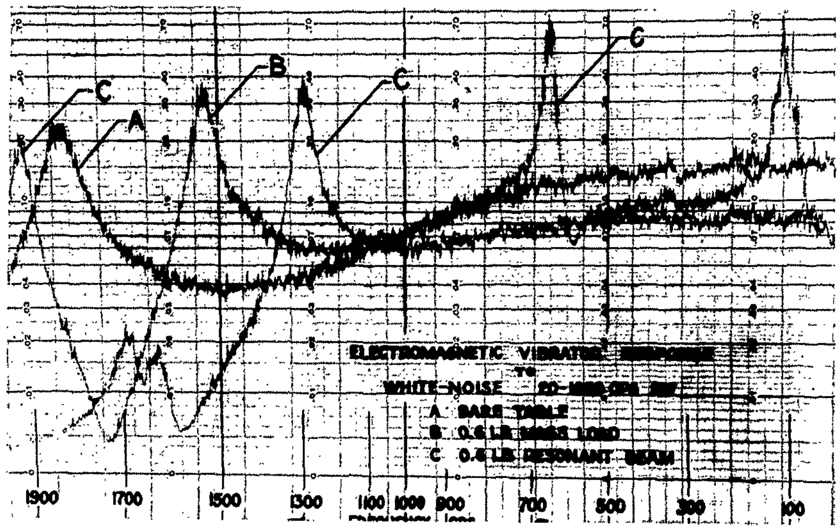

Figure 6

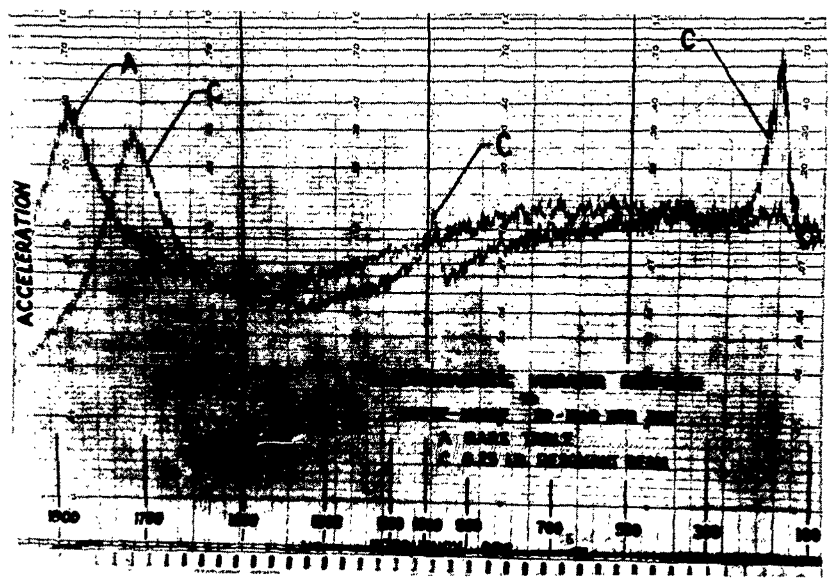

Figure 7

the system. If many resonances of the specimen and shaker each with a different band width are encountered within the test frequency range, then this problem assumes major proportions.

COMPARISON OF SINE-WAVE TO RANDOM VIBRATION

Figure 8 compares the applied spectral density e_f in G per $\sqrt{cps}$ of white noise vibration to the sine wave acceleration, G_s, in g units required to produce an equivalent rms response as defined by the equation:

$$\frac{e_f}{G_s} = \sqrt{\frac{2Q}{\pi f_n}} \qquad (2)^*$$

*Crede, Charles E., "Vibration and Vibration Isolation in Guided Missiles," Barry Controls, Inc; Report No. 257, 25 Nov. 1955

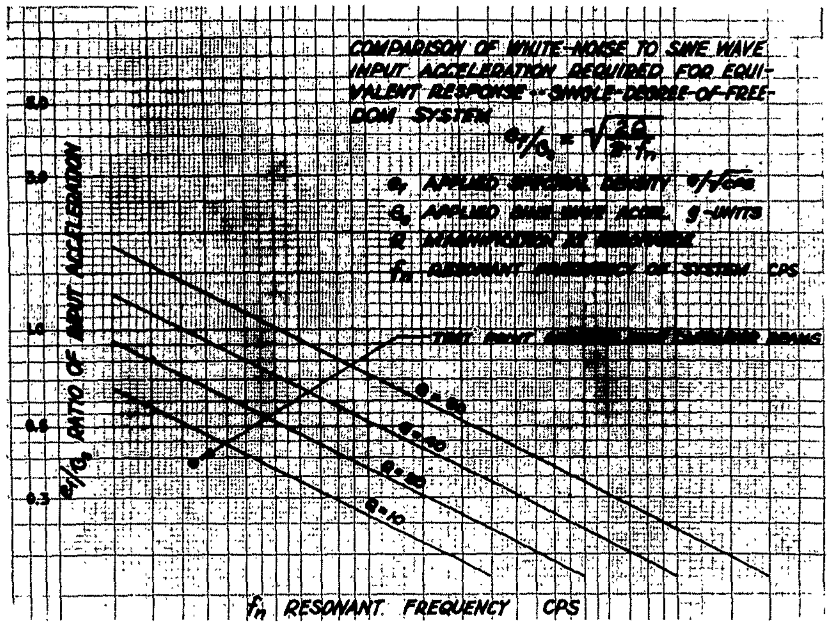

Figure 8

This equation must be qualified in that it applies only to single-degree-of-freedom systems to which a constant spectral density per $\sqrt{\text{cps}}$ random vibration is applied. Also, as far as actual damage potential is concerned, no consideration is given to the various types of statistical amplitude distributions which may be encountered in random vibration. Nevertheless, it is illustrated that the correlation of random to sine wave vibration depends upon the Q of the resonant system and f_n, its resonant frequency. Therefore, it is not possible to arrive at a correlation constant which applies to a complex system with many resonant frequencies and many values of Q.

The single-test point on Figure 8 represents the average ratio of acceleration spectral density to sine-wave acceleration required to cause failure of the test beams where a band width of 20 to 500 cps was used. The Q of these beams as determined experimentally was approximately 10 with a tip double amplitude of 2.5 in. With a tip double amplitude of 1.0 in, the Q is increased to approximately 30.

The poor correlation shown in Figure 8 is not surprising since the beams are not single-degree-of-freedom systems. Also, the fact that the damping of the beams was extremely non-linear in the high amplitude-range may contribute to this apparent discrepancy.

SUMMARY

The foregoing experiments have served to illustrate a few of the rather serious problems connected with applying random vibration in general laboratory work. These experiments have shown two major limitations:

First, to conduct broad-band random vibration tests, facilities with very high acceleration capabilities are required. Therefore, producing random vibration in practical laboratory work may require the development of costly vibration test facilities with greatly increased performance requirements.

Secondly, the responses of vibrators to internal resonances or to resonances of the test item may result in wide variations in the random noise acceleration applied to a test specimen. Although it should be possible to correct the peaks and valleys in the random noise response with filters, experience may show the task to be extremely complex and tedious.

These illustrations of problems connected with random noise testing are by no means the only serious problems. For example, there are questions to be answered such as "How do we compensate vibrator response for rotational motion induced by a dynamic condition of the specimen?" "How certain are we that intolerable

malfunctions observed in the laboratory, which are due to several simultaneous resonances, will actually occur in service?"

The real value of random vibration test techniques actually cannot be appraised until more is known about the true service environment. Investigations must be conducted on the structural conditions in widely scattered regions of missiles. These investigations ought to be of a thorough, comprehensive nature similar to those conducted on aircraft. It is essential that the highest priority be given to these assignments because from these studies and from laboratory experiments, we shall be looking for the answer to the question: "Is it necessary to adopt random vibration testing as a general requirement?"

* * *

THE APPLICATION OF ANALYSIS TECHNIQUES TO LABORATORY TESTING

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The application of analysis techniques to the problem of laboratory testing should help to predict whether a given measured vibration is likely to cause a malfunction in a component. Methods of flight-vibration analysis are reviewed along with the problem of vibration transmission on a simple structure. Consideration is given to the problem of correlation of failures produced by a simulated multi-frequency environment with the failures produced by sine-wave tests.

The goal of the vibration analyses and tests is the development of a given system which will function well during its lifetime in an environment which includes vibration. During this symposium the emphasis is on the comparison of the several schemes of testing systems in the laboratory. It has been suspected, for example, that the sine-wave sweep technique is not adequate, and the question has indeed been raised as to whether it proves anything at all.

A general review is helpful in trying to find some answer to the problem. In the beginning, one presumably starts with a vast amount of recorded data in the form of input accelerations to the system under study. While a complete analysis of these records is not being proposed, it is desirable to describe such an analysis. One can refer to it as a perfect analysis to which all other studies can be compared. In what would it consist? The mathematical description which comes to mind is that of a Fourier Series. The quantities which are needed to describe the series are the amplitudes and the phase angles. The frequencies of the terms correspond to integral multiples of a fundamental frequency. The fundamental frequency is not arbitrary, but depends upon the time interval of the record under examination. If the time interval is 1,000 seconds, the fundamental frequency is 1/1,000 cps. If the input acceleration contains information up to 1,000 cps, the determination of 1,000,000 Fourier amplitudes and 1,000,000 phase angles are required to describe the function. This would be a staggering task.

The perfect analysis of one time interval of 1,000 seconds is not enough. One has to get records for other time intervals. The sum total of Fourier coefficients and phase angles then is available for analytical studies. Bearing in mind that only one input at one point in the system is represented, consider that the other inputs are equal to it in all respects. One then asks about the behavior of an end function of interest. Such an end function is the contact force of an arm on a relay. If the transfer characteristics are completely determined both as to amplitude ratio and phase lag, then the knowledge of the Fourier input amplitudes and phase angles may be employed to frame a series expression for the instantaneous contact force of the arm on the relay.

As a practical matter, no one has even approached such a thorough analysis. What steps are usually taken? The long record is divided into strips. These strips are studied by means of band-pass filters and a quantity known as the power spectral density is derived. The power spectral density is not easy to visualize.

Perhaps the best way to look at the problem is to consider a room full of 50 audio oscillators, each having the same output voltage. Have them all connected so that the outputs add. Suppose that 25 of the oscillators are tuned in frequency increments of 2 cps in a range of 150 to 200 cps. Next suppose that 25 are tuned in increments of 4 cps in the range from 200 to 300 cps.

Let the combined output be used to drive a shaker upon which is mounted a spring-mass system tuned at 175 cps. Assume its damping factor to be such that it responds sharply to forces at frequencies anywhere from 165 to 185 cps. Ten oscillators are thus effective in that 20 cps band, and the combined shaker forces is made up of ten components which are effective in driving the spring-mass system. Then retune the spring-mass system to 250 cps, and leave its damping factor unchanged. The frequencies to which it is sensitive then range from 236 to 264 cps. It is apparent that it is effectively subjected to seven forces. Intuitively, then, one sees that the motions in the first case are greater than they are in the second. The power spectral density is said to be greater in the first case.

The word "power" is somewhat misleading. At times the spectral density refers to acceleration, or to displacement, or any quantity of interest. A short and incomplete discussion is given here in order to introduce a term which may be unfamiliar to many readers.

The power spectral density and a transfer characteristic or transmissibility are needed for analytical studies. The transmissibility, in a general sense, is the ratio of the desired quantity to a known input quantity at a given frequency. A graph of transmissibility versus frequency can be used with a graph of input power spectral density to derive a root mean square value (rms) of the desired quantity.

It is realized that the rms value of any quantity does not convince anyone that the variable is going to stay within satisfactory bounds. How high are the peaks? In order to come to grips with this problem, statisticians have examined various functions of time and have developed theories about one kind called a stationary random function of time. If the function is random, it is not predictable in detail based upon any previous knowledge. If it is stationary, its amplitude distribution pattern for each strip of record seems to be like that of every other strip. Once these characteristics have been established, then the statistician can make remarks to answer the question: How bad can things get for how long a time? In order to take advantage of these techniques, one has to apply certain tests to the given vibration record to classify it as being both stationary and random.

In some cases which the writer has seen, the function has not been stationary. The statistical techniques must be handled with care. Perhaps they do not apply at all. One approach is to study the behavior of electrical analogs to mechanical systems when various measured flight vibration accelerations are played into the analogs. Some interesting uniformities have appeared, such as the fact that the peak response of all the analogs tested never exceeded 4 times the rms value. An appreciation of what an analog does is gained by considering what would have to be done analytically with a record 1,000 seconds long, as mentioned earlier. A million amplitudes and a million phase angles for both the input wave, and for the transfer characteristic would have to be considered.

Raymond Mindlin of Columbia University is to be credited with the suggestion that analogs might be used to advantage in trying to understand flight vibrations. Dr. Robert Mains and R. L. Stallard applied and extended the concept in studies of the Talos missile flight vibration (1). Much information has been gathered concerning the statistical behavior of the response of single-degree-of-freedom analogs. Such information is of value in trying to understand the nature of input vibrations which are not easy to classify as stationary and random.

The foregoing discussion has been given to answer questions about what can be learned about input vibration waves. Let us now consider the problem of a man with a component in need of testing. In order to clarify the picture somewhat, consider a relay. Vibrate it sinusoidally in its vertical plane and monitor an instrument which tells you that the relay has or has not broken contact. Record the acceleration level at each frequency at which the contact is broken. Refer, then, to the curve as a "G to Failure" curve. How can this information be used to reach a decision about the reliability of this one contact in the vertical plane of vibration?

Another set of measurements has to be acquired. Using the actual package, find the vertical acceleration at the point where the relay is to be mounted when the driving sinusoidal vibration at the mounting points is at a constant level, and is slowly varied over the spectrum of frequencies expected to occur in

practice. The ratio of the acceleration at the mount to the acceleration at the input, plotted against frequency, is termed a transfer characteristics, "Y_1" (Figures 1 and 2).

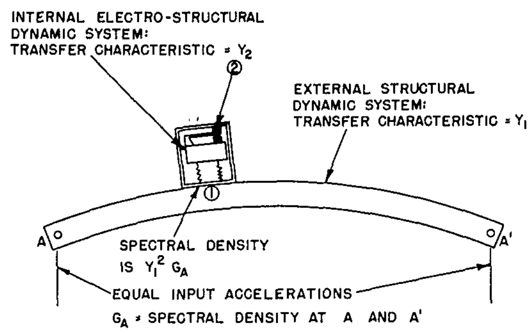

Figure 1 - Internal electro-structural dynamic system

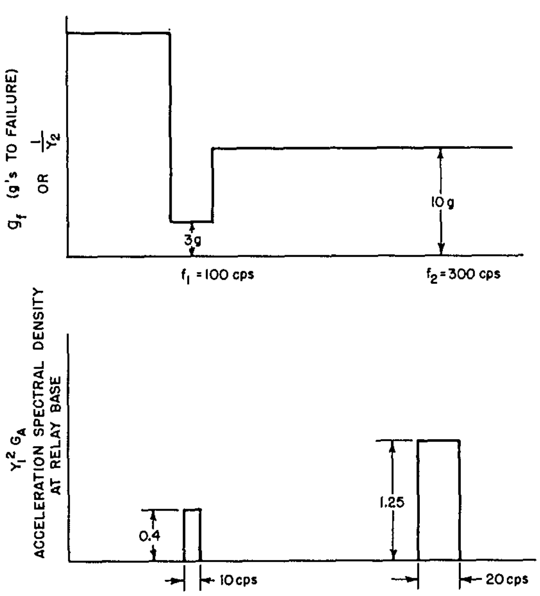

Figure 2

Recall that an acceleration spectral density was derived from flight records. An attempt will be made through a sample problem to show how the spectral density, the transfer characteristic, and the "G to Failure" curves can be used to compute the likelihood of failure.

Assume that the transfer characteristic, "Y_1" and the input acceleration spectral density "G_A" are such as to permit a representation of the excitation spectrum as being made up of only two accelerations, one with a rms amplitude of 2 g at the frequency of 100 cps and the other with a rms amplitude of 5 g at a frequency of 300 cps. Suppose that the rms sinusoidal acceleration required to produce failure is 3 g at 100 cps and 10 g at 300 cps. Refer to Figures 1 and 2.

It is convenient to think of the two exciting vibrations acting separately. If the system is linear, and a 2 g (rms) exciting vibration is available where a 3 g (rms) sinusoidal test vibration is required to produce a failure, then it is natural to think that the device is 2/3 of the way along to trouble. Such a number can be called a "susceptibility ratio". It would be a reasonable estimate provided that the flight wave and the test sine wave had similar amplitude distributions.

It is granted that it is difficult to compare the response amplitude distribution produced by an input wave of varying amplitude at one frequency with the response amplitude distribution produced by an input sinusoidal wave at constant amplitude at that frequency. This is not the only problem. It is difficult to see how the two ratios can be combined. In order to gain an understanding of the problem it is suggested that it is similar to the problem of finding the rms value of one quantity when input and transfer data are given as shown in the following formula taken from Reference 2.

$$\text{rms} = \sqrt{\Sigma\ Y_2^2\ Y_1^2\ \ G_A \Delta f}$$

The application of this formula proceeds as follows: that one value of $\sqrt{Y_1^2\ G_A \Delta f}$ is the 2 g input acceleration, and that the transfer characteristic, "Y_2" is the inverse of the sinusoidal acceleration required for failure at that frequency, namely 1/3.

Likewise, the other value of $\sqrt{Y_1^2\ G_A\, \Delta f}$ is 5 g, and the corresponding value of "Y_2" is 1/10. It is convenient to refer to the end result as the net susceptibility ratio, based upon rms values, or N.S.R. (rms). In this case:

$$\text{N.S.R. (rms)} = \sqrt{\left(\frac{2}{3}\right)^2 + \left(\frac{5}{10}\right)^2} = 0.83$$

The fact that this number is less than unity does not insure safety of the component. It is presently believed that the peak value of the N.S.R. ought to be used. The notion is that in the case of a relay it represents the ratio of the releasing force to the preload force on the contact, and that the instantaneous peak value determines whether or not the contact is broken.

The problem of the peak response of a system having many degrees of freedom is relevant. The peak response of such a system to a complex wave input is likely to be more than 1.414 x the rms value. At present, the only information at hand is that which has been obtained for single-degree-of-freedom systems with a certain damping factor responding to Talos flight vibrations. The greatest measured ratio of peak to rms value was 4. In the absence of any better information, a trial value of the peak net susceptibility ratio is:

$$\text{N.S.R. (Peak)} = \frac{4}{1.414}\sqrt{\left(\frac{2}{3}\right)^2 + \left(\frac{5}{6}\right)^2}$$

$$\text{N.S.R. (Peak)} = 2.35$$

It is concluded that the hypothetical component is unacceptable for use in the given vibration environment, because the net susceptibility ratio of peak values is 2.35 times greater than a failure value of unity.

We are now prepared to ask whether a sine-wave, slow-sweep test establishes any facts about the resistance of a system or a component to a complex vibration. If it is found that a package can withstand a 10 g (rms) test from 0 to 3,000 cps, then how much complex vibration can be endured?

It should be pointed out that a given package might be able to endure much greater accelerations at some frequencies. With the problem as given, however, the most pessimistic view is that at any increased acceleration a malfunction would occur.

This view of the problem establishes the minimum values of the transfer characteristic, "Y_1", and the $Y_2 = (1/gf)$ curves of all the devices within the package. All of the net susceptibility ratios are at least less than:

$$\text{N.S.R. (Peak values)} \leq \frac{4}{1.414}\sqrt{\Sigma\left(\frac{1}{10}\right)^2 G_A \Delta f}.$$

Choosing a value of N.S.R. of unity results in a determination for $\sqrt{G_A \Delta f}$ a quantity which is an allowable rms input acceleration. In this example, the allowable value of 3.5 g is predicated upon the empirical factor of 4 which may have to be changed when more information becomes available.

The next most conservative thing to do is to establish a "G to Failure" curve for the package as a whole, and once more apply the formula. This, too is conservative, because in one frequency band, one component may be sensitive, while in another frequency band, another may be sensitive. If one uses the minima of all the "G's to Failure" curves, he will be conservative in his calculation of a net susceptibility ratio. This discussion has assumed that for one reason or another it is inconvenient to obtain detailed "G to Failure" curves for each component.

If the "G to Failure" curve is available for each elementary function of each component, and the transfer function of the structure is available, then a net susceptibility ratio may be computed for each of the functions. If it is learned from many tests that these ratios are meaningful, then this ability to predict is the first step toward control. The design of a structure so that all of the ratios will be less than 1, is an important task which will not be discussed in detail. It is helpful to list the transfer characteristic variables for a structure. The transfer characteristic of a linear structure is made up of contributions from an infinite number of vibration modes. The degree to which each mode contributes to the total displacement is influenced by five factors; namely: the location of the point, the mode shape, the mass distribution, the natural frequency, and the damping factor.

For example, the transfer characteristic for a simple pin-ended beam is given by the following expression, taken from Reference 2:

$$Y_1^2 = \left[\frac{\text{Output Acceleration}}{\text{Input Acceleration}}\right]^2 =$$

$$\left[1 + \sum_{k=1,3,5}^{\infty}\left(\frac{\omega}{\omega_k}\right)^2 D_k A_k H_k\right]^2 + \left[\sum_{k=1,3,5}^{\infty}\left(\frac{\omega}{\omega_k}\right)^2 D_k A_k V_k\right]^2$$

where

ω = driving frequency,

ω_k = resonant frequency of k'th mode,

$\omega_k = \frac{\pi^2 k^2}{L^2}\sqrt{\frac{EIL}{M}}$,

M = Mass of beam,

L = Length of beam,

EI = Bending stiffness,

η = Damping parameter,

$A_k = \frac{4}{k\pi}$

$$D_k = \sin k \frac{\pi X}{\ell},$$

$$H_k = \frac{1 - \left(\frac{\omega}{\omega_k}\right)^2}{\left[1 - \left(\frac{\omega}{\omega_k}\right)^2\right]^2 + \eta_k^2},$$

$$V_k = \frac{\eta_k}{\left[1 - \left(\frac{\omega}{\omega_k}\right)^2\right]^2 + \eta_k^2}.$$

An important consideration is that of the natural frequency. Both intuition and analysis agree in requiring that the natural frequency of any mode should be separated from the minimum points in a "G to Failure" curve.

The mechanisms by which a measured vibration wave may produce a malfunction involve careful analysis of all the input vibrations, bearing in mind that they do not fall into convenient classifications. A useful tool in understanding vibrations is the effect produced upon mechanical systems or analogs of mechanical systems. The transmission of vibrations in any practical structure is hard to compute, but insight into measured values and techniques for changing them are provided by linear normal-mode theory.

The notions of power spectral density, transmissibility, and net susceptibility ratio have been described to show a process by which an estimate of the component reliability can be computed. The validity and utility of the process have not yet been demonstrated. It is hoped that the results of future tests and field experience will be in accord with predictions provided by the net susceptibility ratio of peak values.

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DISCUSSION

Dr. McCalley, General Electric Co.: Could you tell me if this other paper you referred to with the derivations in it is available? Is it possible to get a copy?

Kearns: Well, there are a number of derivations on the market and each time I come up with one of them, as here, I have to refer to some other paper. Parseval's Theorem is known and used, but it was not known to me when I needed it. I got the derivation from Rice's paper. Mr. S. O. Rice wrote a rather distinguished paper on the mathematical analysis of random noise which is always good to include for a reference. I could tell you to go to that one, but I have a derivation, too, in APL report CF-2422. It is unclassified. What I really get there is a correlation between the coefficient of the input wave and the power spectral density. It shows you how to get a rms value of the quantity. Knowing the power spectral density of the transmissibility curve is just one step further and can be done exactly the same way.

* * *

THE SELECTION AND PERFORMANCE OF SINGLE-FREQUENCY SWEEP VIBRATION TESTS

Allen J. Curtis, Hughes Aircraft Co.

Characteristics of response of a system to actual environments and to typical specification tests are compared for the purpose of selecting appropriate tests. The use of sweep tests for transient fatigue and production testing is discussed, with emphasis on the selection of a sweep rate for each. The relationship of desired tests to vibration table capabilities is mentioned briefly.

Many of the vibration tests required in Government specifications are of two types. One type calls for a constant displacement and/or constant acceleration input to the test object while the frequency is varied continuously over a given frequency range or bandwidth in a prescribed time. The other type requires a constant input at the resonant frequency or frequencies of the test object, for a given time.

It is the purpose of this paper to present some observations on the adequacy of the first type of test and the manner in which it should be performed, and the compromises that must be made when practical considerations of test equipment, etc., are taken into account.

Let us first consider the nature of expected shock and vibration environments for military equipment, particularly, airborne equipment, such as fire-control systems, missiles, etc. Available data indicate that the vibration encountered in flight is essentially random in nature, that is to say, the excitation has harmonic content over the whole frequency spectrum, and the amplitude and phase at each frequency is varying in a random manner with time, and is normally distributed. In addition to this random vibration or noise, some steady-state excitation at one or more frequencies may exist. Finally shocks, or short-duration transients whose level is significantly higher than the accompanying random and steady-state vibration, will occur from time to time, due to take-off, landing, gusts, etc. For a missile, there is also the launching shock from the rocket motor firing, which is probably the most severe shock that it will encounter.

One now asks "How do the two types of vibration tests previously mentioned simulate this type of environment, or guarantee that equipment which has passed such tests will perform efficiently and reliably in the field?" It is apparent that it is not too important for a test to simulate exactly an environment providing it adequately assures performance and reliability. However, since there is only a remote chance that a humidity test will assure performance in a vibration environment, there will tend to be a definite correlation between the environment and test characteristics if the objective is to be attained.

If the similarities between the responses of systems to the two types of vibration tests and the environment are examined, the selection of the appropriate test is made more easily.

Although the response of systems to random vibration has yet to be thoroughly explored, it

is known that, in general, the response will be at the natural frequencies of the systems with randomly varying amplitude. The response of systems to steady-state excitation is well-defined, needing no further comment, and the response of systems to shock will be at the natural frequencies and may be defined by shock spectra.

Now let us look at the two types of vibration tests. Resonance tests will of course produce a response at the excitation or natural frequency at constant amplitude. Thus resonance testing will simulate the response of systems to steady-state excitation which coincides closely with a natural frequency, and may possibly be used to simulate the response to random vibration statistically. The single frequency sweep, with which we are most concerned here, will excite the resonant frequencies, one at a time, for short periods. In other words, unless the sweep-rate, or rate-of-change of frequency is extremely small, a sweep-test can be considered to be similar to a shock test, except that the natural frequencies and modes are excited progressively during the test instead of simultaneously as in a shock test.

Thus a sweep test does not appear to be an adequate simulation of either a random or steady-state vibratory environment. However, except that the natural frequencies are excited one at a time, it does simulate the transient environment. Accepting this, it is then possible to arrive at the correct input level for a sweep test as follows. The severity of transient excitation can only be defined in terms of the response of a system, and the shock spectrum is commonly used for this. Thus the level of the sweep test should be such that the response of the test object is equivalent to its response in the transient environment, and should be specified in terms of response instead of input.

Of course, it must be appreciated that the foregoing remarks apply to those sweep tests of fairly short duration, which are not intended to be used as fatigue tests. It is felt that fatigue tests are better conducted as random or resonance tests when the significant modes are known. If the resonances are not known, then sweep tests may have to be employed.

Thus far then, for type or proof-of-design environmental tests, a sweep test is satisfactory for simulating transient environments. There is, in addition, one more important use of this type of test, namely, assurance or proof-of-workmanship tests. For this type of test, by definition, the test object is of proven design, and it is desired only to assure oneself that the units being produced are of satisfactory quality. In general, the unit or system will be operated during the test and its performance monitored. Of course, a shock test may be used for the same purpose but there are some advantages to using a sweep test. For example, it is generally easier to monitor the performance and to discover a malfunction in a sweep test than in a shock test. The choice of level for such a test is somewhat arbitrary, and not well defined. Certainly one would expect the level to be less than that used in a type test, but of course it must be high enough to discover the major percentage of production deficiencies without damaging the unit or reducing its operating life. The adequacy of the test can be determined only from the subsequent field performance and reliability of the units.

Having briefly discussed the selection of a single-frequency sweep test, let us briefly turn our attention to the performance of such a test. First assume that conditions of the test regarding frequency range, input to/or response of some point on the test object and total sweep-time have already been specified. Presumably then, it is known exactly how to conduct the test. Yet there have been many instances where tests on the same unit to the same test specifications have produced very different results. There are many possible reasons—with which we are all familiar—that can cause these discrepancies. In the following remarks, some of the effects of the sweep-rate will be discussed.

First, acknowledgment must be made to Maurice Gertel of the Barry Corp., who presented much of the same material in two recent reports (1,2) to WADC. It is felt that repetition here, with some extension by the author, will lend completeness to this presentation.

Now what are the effects of sweep rate on the response of a test object? Certainly if the rate is low enough, the response of each frequency will be the steady-state response to input at that frequency. On the other hand, if the sweep-rate is high enough, the response is not steady-state, and resonances will not build up to their full values. If they did, numerous pieces of rotating machinery would not be operating above the first critical speed of the shaft. Between these two extremes then, lies the sweep-rate which, it is hoped, will fully excite all resonances.

Depending on the purpose of the test, one may also wish to control the number of cycles or time at high amplification. For example,

simulation of transients would require an equal number of cycles, since natural modes with equal Q's will decay in an equal number of cycles. Similarly, a fatigue life-test would dictate an equal number of cycles at each resonance. However, for a service life-test, equal time at high amplification may be more desirable. The appropriate sweep rate will be different for each requirement, and may be determined as outlined in the following paragraphs.

The response of a system to harmonic excitation of variable frequency is not usually amenable to analytical solution. However, F. M. Lewis (3) has obtained a solution for the response of a single-degree-of-freedom system to a vibratory force of constant amplitude with frequency changing at a constant rate. Figure 1 is a reproduction of Figure 2 of Lewis's paper showing the response of a system with 5 percent critical damping for various sweep rates. From the curves presented in his paper, the rate required for the peak amplitude to build up to a given percentage of the steady state value at the resonant frequency may be determined for various amounts of damping. These are not exactly the conditions present in a sweep test, but if the instantaneous sweep rate at each frequency is no greater than would be required to excite a single-degree-of-freedom system to a given percentage of the resonant amplitude, it may then be assumed that approximately the same percentage build-up will occur at each resonance of a multi-degree-of-freedom system. However, for modes with nearly equal natural frequencies, this assumption may no longer be as valid.

For the single-degree-of-freedom system, if the frequency rate is determined from:

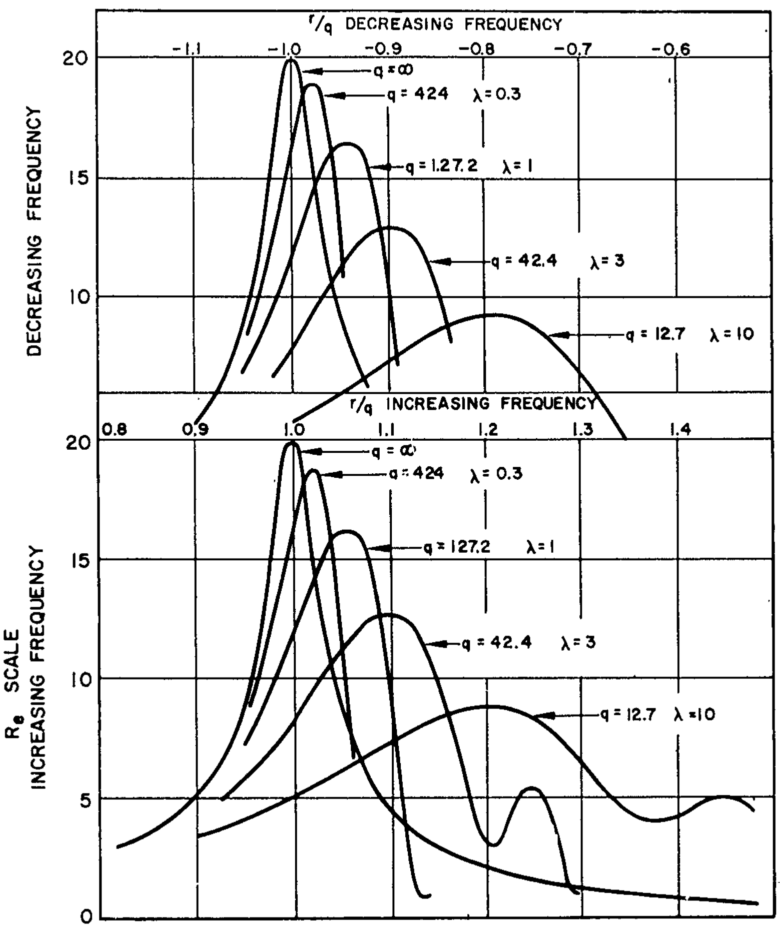

Figure 1 - Response of single-degree-of-freedom with 5 percent critical damping

$$h = \frac{f_o^2}{R} \qquad (1)$$

where

h = rate of change of frequency (cps/sec)

f_o = natural frequency of system (cps)

R = a constant (equivalent to Lewis's q)

The maximum response will be a constant percentage of the steady-state resonant amplitude for any natural frequency. Examination of the curves for several values of damping and R reveals that Equation (1) may be expressed as:

$$h = \frac{4\zeta^2 f_o^2}{k} = \frac{f_o^2}{kQ^2} = \frac{B^2}{k} \qquad (2)$$

where

$100\,\zeta$ = percent critical damping of system,

k = a constant,

Q = maximum amplification factor or transmissibility of system for $\zeta \leq 0.15$,

B = bandwidth of resonance.

If R is large enough to allow the peak amplification to reach, say at least $0.90\,Q$, then the number of cycles above $0.70\,Q$ or $0.80\,Q$ will be constant for varying f_o, and approximately proportional to R for the same damping. In addition, for the same percentage build-up for varying damping [equal k in Equation (2)], the number of cycles above a certain level will be proportional to Q. Combining the last two variations yields the result that for equal R the number of cycles above a given level will be inversely proportional to Q, or proportional to bandwidth, as might be expected.

The response of multi-degree-of-freedom systems can be controlled by using the above relations to determine the instantaneous frequency rate.

If the instantaneous frequency rate is determined from Equation (2) with $k = 1$, each resonance will be excited to approximately 95 percent of maximum amplification. The number of cycles above 70 percent of maximum amplification will be approximately equal to the bandwidth of the mode. However, since each resonance of the system is likely to have a different Q, probably unknown, the rate appropriate to the highest Q should be used, thus subjecting the lower Q modes to more cycles at high amplification.

Figure 2 shows the rate of change of frequency as a function of frequency for $k = 1$ and various damping values.

Equation (1) may be integrated to determine the time to sweep from f_1 to f_2 to produce a desired number of cycles (N) above a given percentage amplification as follows:

$$\begin{aligned} T_N &= \frac{N}{N_R}\int_{f_1}^{f_2}\frac{df}{h} = \frac{N}{N_R}\int_{f_1}^{f_2}\frac{R\,df}{f^2} \\ &= \frac{N\,R}{N_R\,f_1}\left(1 - \frac{f_1}{f_2}\right) = \frac{N\,k\,Q^2}{N_R\,f_1}\left(1 - \frac{f_1}{f_2}\right) \end{aligned} \qquad (3)$$

where N_R is the number of cycles for a given R, which may be determined from the curves of Reference 3.

For amplification above 70 percent, N_R becomes $k\,Q$, so that

$$T_N = \frac{N\,Q}{f_1}\left(1 - \frac{f_1}{f_2}\right) \qquad (4)$$

which is plotted in Figure 3.

Now consider that the time for which amplification above a desired level occurs is to be held constant for each mode. Evidently the time to sweep through the bandwidth is now a constant. Thus the frequency-rate is expressed by the relation

$$\begin{aligned} h &= \frac{B}{t_s} \\ &= \frac{f}{Q\,t_s} \end{aligned} \qquad (5)$$

where t_s is the significant time at each resonance, and the resulting h is small enough to permit adequate response of the system. The time for a sweep from f_1 to f_2 may be found by integration to be

$$T_s = \int_{f_1}^{f_2}\frac{df}{h} = \int_{f_1}^{f_2} Q\,t_s\frac{df}{f} = Q\,t_s\,\ln\frac{f_2}{f_1}, \qquad (6)$$

which is plotted in Figure 4.

Again, when resonances of varying Q's are present, the significant test time is inversely

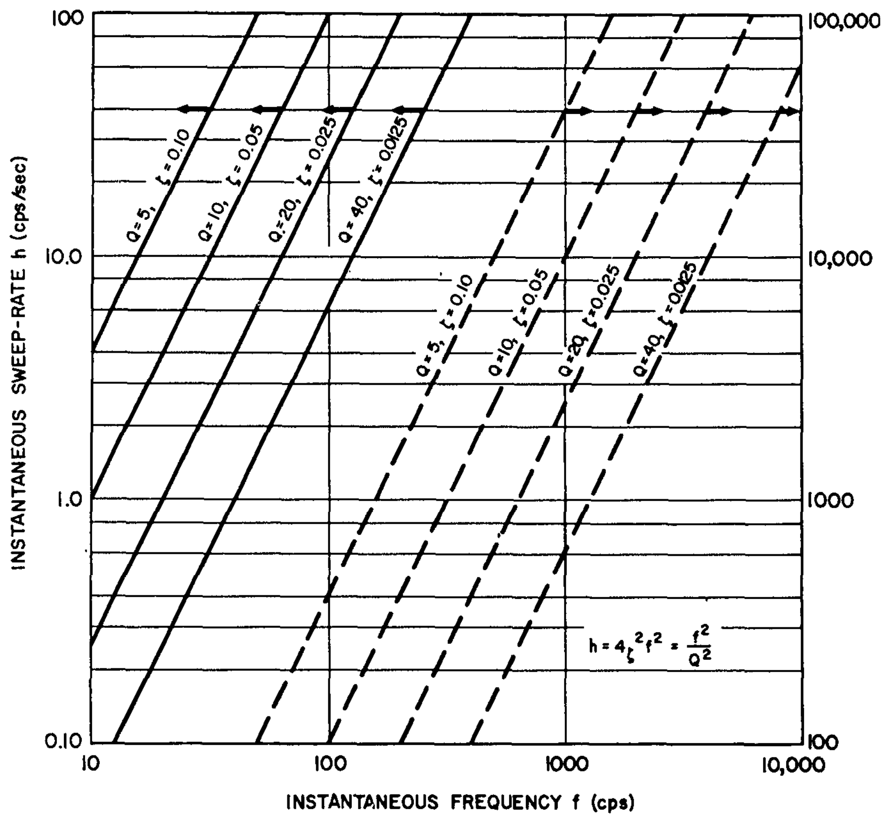

Figure 2

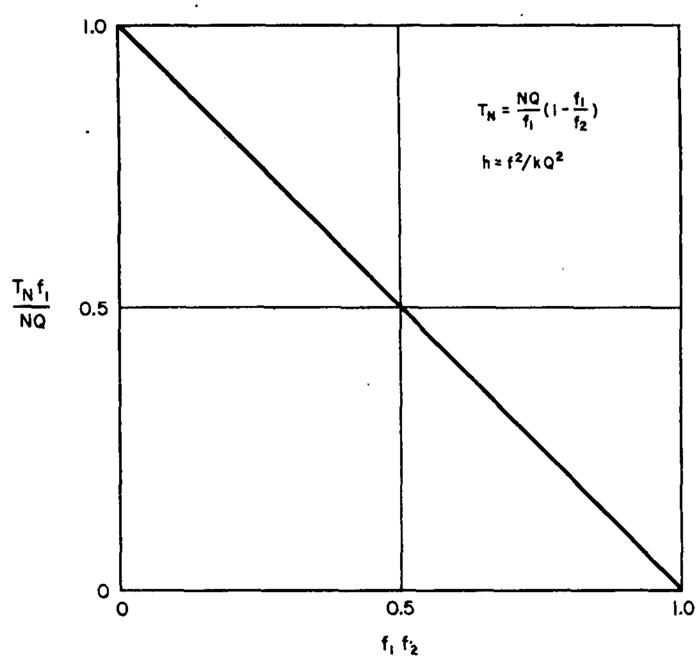

Figure 3 - Time T_n to sweep from frequency f_1 to f_2 with N cycles above 0.70 Q at resonance

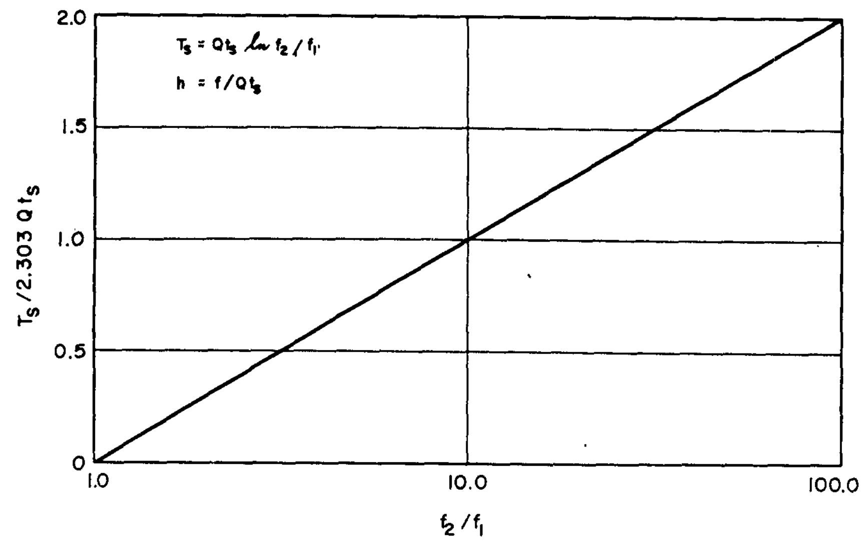

Figure 4 - Time (T_s) to sweep from frequency f_1 to f_2 with significant test time t_s at resonance

proportional to the corresponding Q, and the sweep time must be based on the highest Q for adequate testing. The number of sweeps or total test time is now based on the given operating life.

It is now necessary to relate the preceding comments to capabilities of vibration tables presently available. It is evident that there is a limit to the rate at which the output frequency of a particular vibration table, loaded with a particular test object, may be changed if waveform is to be preserved and if amplitude is to be controlled to the desired accuracy. If a test on a system is to be performed at most a few times and consists of one or two frequency sweeps it may be satisfactory to control amplitude and frequency manually.

However, long-duration fatigue tests or oft-repeated production tests are not economically possible with manual control. Automatic frequency control for linear and logarithmic frequency sweeps is available. It is seen that the logarithmic sweep, providing it is not too rapid, corresponds to the frequency rate of Equation (5), and thus gives a test with equal significant time at each resonance for a service life test. At present, to the writer's knowledge, a control system to produce an equal number of cycles at each resonance for a transient or fatigue life test is not available, although it certainly appears desirable to have this capability.

Little has thus far been mentioned regarding excitation amplitudes of sweep tests. For some tests, a constant displacement or constant acceleration over some frequency band, monitored either on the table or on the test object, will be satisfactory. Automatic amplitude control for such a test, providing the sweep rate is not excessive, is presently available. However, a system with which the amplitude or acceleration could be programmed as a function of frequency appears highly desirable.

To summarize, it is felt that single-frequency sweep vibration tests are often applied without due regard to the environment which they are intended to simulate. In addition, the manner in which they are performed requires closer control of sweep rate if consistent results amenable to correct interpretation are to be obtained.

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2. Crede, C. E., Gertel, M., and Cavanaugh, R. D., "Establishing Vibration and Shock Tests for Airborne Electronic Equipment," WADC Technical Report 59-272, June 1954

3. Lewis, F. M., "Vibration During Acceleration Through a Critical Speed," Trans. ASME 1932

DISCUSSION

Mills, Canadian Westinghouse: I have two questions. One: what is the journal with the F. S. Lewis article? Two: according to an article, I believe it was by Chang Bonart, a logarithmic frequency sweep rate of 45 seconds per decade will result in deflections which are 99 percent of the deflections you would have obtained if you had stayed on one frequency with a maximum Q. Does that figure tie up with your F. S. Lewis article?

Curtis: I am not familiar with the reference you cited and I would certainly like to look it up myself. The Lewis article I referred to is "Vibration During Acceleration Through A Critical Speed," and it appeared in Transactions, ASME in 1932.

Nitchie, ORL, Penn. State Univ.: Do the curves on Figure 1 apply to both directions of sweep?

Curtis: The top curve is for decreasing frequency, and the bottom one for increasing frequency. When the frequency is increasing the peak amplification will occur at higher than the natural frequency. Conversely, if the frequency is decreasing, the peak amplification occurs when the excitation frequency is lower than the natural frequency. In effect, in the top picture time is going to the left, and in the bottom curves time is going to the right.

Fine, NOL, Corona: You didn't mention anything about subharmonics. How about that logarithmic time with respect to subharmonics? Is the curve still linear?

Curtis: Let's put it this way: Lewis's paper is based on a linear single-degree-of-freedom system, and it has just been postulated that if you have a multi-degree-of-freedom system with resonances and modes which are not too strongly coupled together, then you can apply his results to the multi-degree-of-freedom system. So implicitly we have assumed that it is still a linear system. What would happen with subharmonics, I certainly would not wish to comment on.

Steiner, Sperry Gyroscope: I was wondering if there was any particular reason why most of the vibrator manufacturers are sweeping linearly rather than logarithmically?

Curtis: Well, there are some of them in the audience that I recognize, maybe they would like to answer the question. I have no ideas myself.

Zimmer, Calidyne Co.: The logarithmic sweeps are currently available both for sweeping with oscillators or with motor generator sets.

Curtis: Excuse me. I am not sure if Calidyne puts out equipment that sweeps linearly, but why do manufacturers do this? Could you answer that question?

Zimmer: As far as sweeping is concerned, it was, in the early stages fairly simple to just hook up a little motor to an oscillator dial and turn the motor over at a constant speed. Your rate then depended entirely upon what the oscillator was set up for. In the case of rotary driving equipment, this was dependent entirely upon the particular motor control, which was in most cases, I believe, the Ward-Leonard control.

C. Morrow, Ramo-Wooldridge Corp.: I have just one comment on this paper. You discussed this morning the single frequency sweep, and the relationship between amplitude and Q. Assuming that the sweep rate is quite low, as you have pointed out, we still have a certain amount of sweep rate to play around with. One thing that can be done in getting a little better

approximation to a single-frequency equivalent is to pick some maximum Q, and then sweep at a rate that is fast enough to reduce the response at that maximum Q to essentially what you get for low Q's. In other words—percentage response. According to this approach you would get a higher than desired response for the middle range, but the over-all approximation would be better. The only reason that I have not pursued this line of thought very far is that I haven't been sure you could control a motor-generator set to the precision required here. In other words, if the motor-generator moves at-all spasmodically then every so often you do get the full response that you are trying to avoid. If you have an oscillator feeding an amplifier, you can probably control this a little bit better, but then you will already have the makings of a complex wave system. But it may be that the motor generator sets can do a little better job than I had assumed.

Curtis: Well, I quite agree with you, Dr. Morrow. This could be used, but we have never tried to do so. My plug was to have us at least consider sweep rate a little more than we, or at least I, have done in the past.

Whiteley, Naval Ordnance Lab., Corona: From a reliability standpoint, I am interested in one aspect of this problem, which I haven't heard mentioned yet. That is the definition of a test or a sweep frequency which would be done at some time-rate as a standard, and starting out with some low amplitude which would not be destructive. I think the present trend, reliability-wise, is such that we need to know how good a piece of equipment is, and not just that it passes a specification. It would seem the right way to achieve this answer is to provide a test which induces failures. I think that there is, in the current literature, a large amount of information which indicates the number of failures which should be induced in order to determine the mean time to failure, and that production control tests on the equipment should be based on this determination of the mean time to failure. This would confirm, within certain bounds, what the designer had in mind when he designed the equipment. I'd like to know if there is anyone in the audience who has been thinking along this particular line, of a standardized sweep-frequency test which can be generated to evaluate how good the piece of equipment is?

Nankey, ARDE Assoc.: I would like to say that in this concept of testing to destruction it is a good idea to form a destruction envelope, but it still leaves wide open the question of the equivalence of a single-frequency sweep and actual random environment.

Unholtz, MB Mfg. Co.: I'd like to say a word on linear sweep rates. I am quite sure that some of the folks that have done a great deal of environmental testing feel that the linear sweep rate is justified on the basis that if you have sinusoidal excitations, the number of cycles an aircraft would experience would depend primarily upon the frequency, because the lower frequency excitations would have fewer cycles in, say, a two-hour flight than the higher frequency excitations. Therefore the linear rate tends to give more cycles in the higher frequencies and less cycles in the lower frequencies rather than an equal number at any resonance or an equal degree of build-up at any resonance.

Curtis: If I may answer Mr. Unholtz, the logarithmic sweep is the one which gives an equal time at each resonance, and the thesis here is that if you are thinking of a service life test where the environment is given as so many hours, then you wish to spend an equal amount of time at each resonance. This is the logarithmic sweep.

In the parabolic sweep, that is one name for it, the sweep rate is proportional to frequency squared and this gives you an equal number of cycles. But with the linear sweep then you are one degree further the other way. The sort of cycle you are simulating assumes that an item has a longer life at the lower frequencies or that the life is inversely proportional to the frequency. That is why I said I didn't know why we used linear sweeps.

Now, of course, implicit in this is the idea that the only stresses that matter are the ones that occur at resonance. If it is just amplification of 1 or 1.2 which is going to be damaging, then maybe you're right. But I think most of us agree that it is the resonances, resonant amplitudes, or amplification at resonance which are the damaging criteria.

Granick, WADC: I'd like to say one more word about linear versus logarithmic sweeping. Most of us usually make the assumption that damping is not a function of resonant frequency. That is to say, we presume that regardless of what the resonant frequency is, the Q of the curve is the same. Now experience shows us that this is not exactly true. At the higher resonant frequencies we find the bandwidth to be smaller, the Q's to be higher, and therefore, we feel that a linear cycling rate is more indicative of an equal or proportionate number of cycles.

Curtis: I believe I mentioned that the one big trouble is to know what your Q's are and to

allow for them. However, if you control your sweep rate properly you are not going to get higher amplification. If you use a sweep rate which is no greater than necessary to excite the highest Q resonance, you will get up to 95 percent amplification so that slowing down even further is not going to help. My own feeling is that these high Q modes occur when we deal with materials which are hardened, such as the internal elements of tubes. You can make a very low frequency, low Q device but if you spring-temper the steel, you can get a very high Q. I think the reason for low Q's and low frequencies may be found in the way in which we use the materials. It is my own personal opinion that it is not a function of frequency but a function of the materials.

* * *

DEVELOPMENT AND USE OF A SHAPING NETWORK FOR COMPLEX-WAVE TESTING

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The design of a set of networks to compensate for the transfer characteristics of an amplifier-vibrator combination is discussed and calibration techniques for the system are reviewed. The use of the system is discussed and examples are given of the types of spectra obtainable. A test program utilizing the system is reviewed and future plans are outlined.

INTRODUCTION

The time-honored method of running vibration tests has been with a single degree of frequency sine wave. The military specifications have been, and still are based upon the sine wave for both type and production tests. Either the dwell test, where a single frequency is held for a prescribed time, or a sweep test, where a frequency spectrum is traversed in a prescribed time, is specified. The trend has been to require a higher frequency as an upper limit.

Recently, in the missile field in particular, there has been an increasing awareness of the fact that the sine wave test is, at best, a poor simulation of the actual vibration environment. The telemetered data from flight indicate that the missile environment is a complex wave containing many frequencies which appears to be somewhat random in amplitude. It is, therefore, rather obvious that the profession begin to think in terms of complex-wave or noise testing in addition to sine-wave testing. Several complex-wave systems have been developed and are in use at various organizations in the country. In this paper we will describe a system which was developed at the Applied Physics Laboratory to enable components and missile systems to be subjected to a noise-type vibration test.

Although complex-wave testing is being used more widely, we should realize at the outset that while this is a much closer simulation to the actual environment of a missile there is still a lot that can be accomplished with sine-wave tests. The determination of resonances and transmissibility curves will always be important in the development and evaluation of designs.

Let us assume that we could subject a black box, containing some electromechanical device, to the exact vibration environment which it would receive in use. Assume further that its output function is out of tolerance during this test. It would then be necessary to trouble shoot by means of sine-wave vibration to repair the device. However, complex-wave testing will be valuable to indicate how a finished component lives in the environment to which it will be subjected. There is still much to be learned with regard to component fragility when subjected to complex-wave vibration. Studies of this subject should be the immediate application of any system. The need for a realistic

quality-assurance test seems to be a further application. It appears then, that complex-wave testing will satisfy as a production test, while sine-wave tests will still be a valuable tool and indeed may be the most advantageous way to perform a type test.

THE OVER-ALL SYSTEM

In order to reproduce faithfully a flight vibration environment, one must generate a complex-wave vibration of considerable magnitude. This means that the fidelity of the shaker system must be good and that a relatively large amount of power must be available. The latter problem is solved simply by providing a large amplifier. Our system uses a 15-kw amplifier and even larger ones are in use.

The commercial electromagnetic vibrators generally have a response curve which is definitely not flat and which varies with the mass on the vibration table. There are three basic approaches to correct this problem.

- The first is to build an amplifier-shaker combination with a flat response up to a reasonably high (30 kc) frequency.
- The second is to build a servo or feed-back system which would apply automatic correction.
- The third approach would be to shape the spectrum.

Ultimately, one of the first two approaches might provide a better system, particularly for production tests, but we felt that the construction of shaping networks offered the best solution at this time. These networks are placed in front of the amplifier, between the signal source and the amplifier, and can be adjusted to give the type of spectrum required. In certain cases we would want the spectrum to be as nearly flat as possible, while in other cases we would want to shape the spectrum to some predetermined curve. This allows us then a great deal of freedom in the type of tests that can be conducted. Since our interest is to explore the effects of various waves upon components as well as test-missile systems, this flexibility was important.

The over-all system is shown in block diagram in Figure 1. A signal source, which could be an oscillator or flight tape or a noise generator, feeds into a set of shaping networks, from there into an amplifier, and then into the shaker. The calibration system, which is a sonic analizer, is also shown. Figure 2 is a photograph of the actual equipment, showing a 15-kw amplifier with its control panel, a 3,500-lb force shaker, the sonic analyzer, and shaping networks which were built at APL. The networks are also usable with our smaller shakers and amplifiers so that appreciable flexibility is obtained. Before discussing their use, we will describe the construction of the shaping networks.

THE SHAPING NETWORKS

The networks consist of a master unit and 20 separate filters, each one of which can produce a notch or a peak of variable amplitude at any frequency from 20 to 2,000 cycles. We might

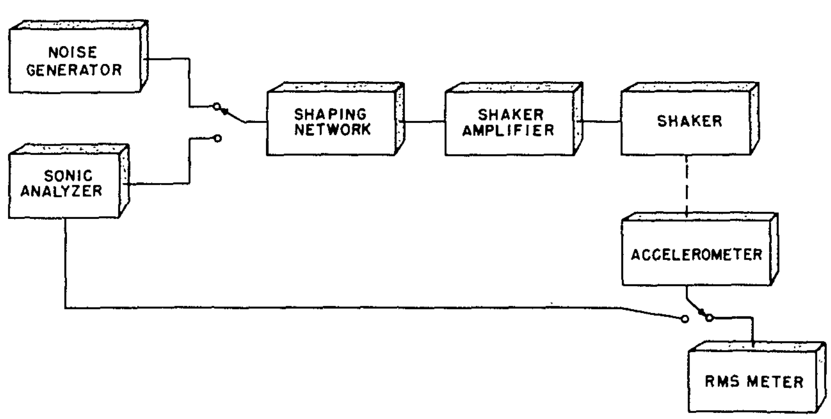

Figure 1 - Block diagram of complex-wave equipment

Figure 2 - Complex-wave vibration test equipment

point out here that the 2,000-cycle upper limit is presently the limit of our telemetered data, but it can be increased. The master unit accepts the input signal and feeds the voltage into a cathode follower stage so that the high-impedance input is converted to a low-impedance output which is used to drive any or all of the channels in series or parallel. A master gain control is also provided on this unit, which is usable to set the input operating level. Each of the channels is, of course, provided with its own gain control which sets the amplitude of that particular channel.

The basic part of the system is a bridged T network, the schematic of which is shown in Figure 3. This frequency-elimination RC network has a low hum pickup in electromagnetic fields, high-rejection characteristics over a broad range, and a high Q at low frequencies. In addition, it seemed that the circuit could be built with readily available commercial components. The circuit values were chosen as:

$$R_1 = 2\,R_o,$$

$$C_1 = C_2 = \frac{1}{2}\,C_o,$$

$$R_1 = \frac{1}{2\pi\, f\, C_1}.$$

The maximum attenuation is obtained from this circuit when the frequency is equal to $1/2\pi\ R_1\ C_1$. The high-frequency limits of the unit are set by the 2 4K and 1 2K resistors. A three-gang potentiometer which adjusts R_1, R_2, and R_o provides the frequency control. A width control is also provided in the network. The type of notch that can be obtained from a single filter is indicated in Figure 4.

Each filter is capable of being put into a notch or peak position and has three controls. The frequency control, of course, selects the point

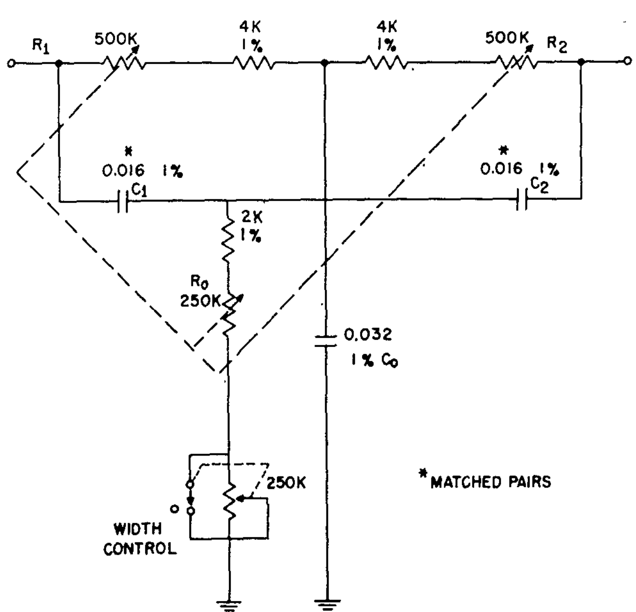

Figure 3 - Bridged T filter schematic

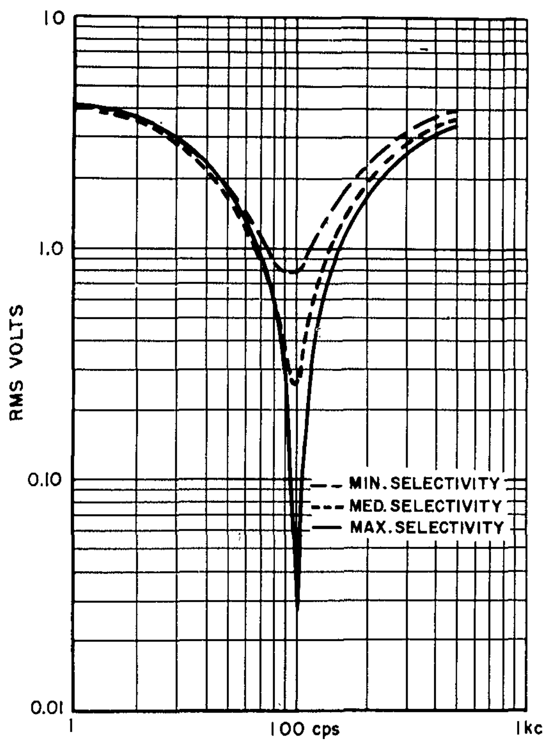

Figure 4 - Bridged T filter notch characteristics

at which either a peak or a notch is desired. The gain control selects the amplitude of the peak or the notch, and the width of either the peak or notch is adjustable. The 20 channels were built as five units, each of which contained four filters. Each of the filters was designed to be capable of inserting a notch or a peak characteristic of 20 to 1 ratio or better in the range of 20 to 2,000 cycles. Any of the filters can be switched in or out so as to provide a peak, notch, or it can be left flat, i.e., no correction. A series circuit is provided which allows the filters to be cascaded for rejection or for the addition of individual frequencies. Matching transformers are used between each unit of four filters to compensate the attenuation of high frequencies due to shielded cables. This feature enables the equipment to be used in high-intensity electrostatic and magnetic fields such as are encountered in environmental testing.

A close-up of the master unit panel is shown in Figure 5. This unit contains a filter unit having the center controls in addition to the cathode follower stage and monitor. The frequency control at the top adjusts a three gang potentiometer. The reader will note a tag "peak 40 cycles" above this control. We found that it was not possible to get maximum gain at all frequencies with one filter due to tracking errors. It was then necessary to pad the circuit to obtain a frequency at which a gain of 100 to 1 could be realized, with lesser gains at other frequencies. On the filter shown, maximum gain is obtained at 40 cps.

With a choice of a series or parallel adjustment of 20 units, and three controls on each unit, it is obvious that correction of the spectrum is somewhat time consuming. As discussed

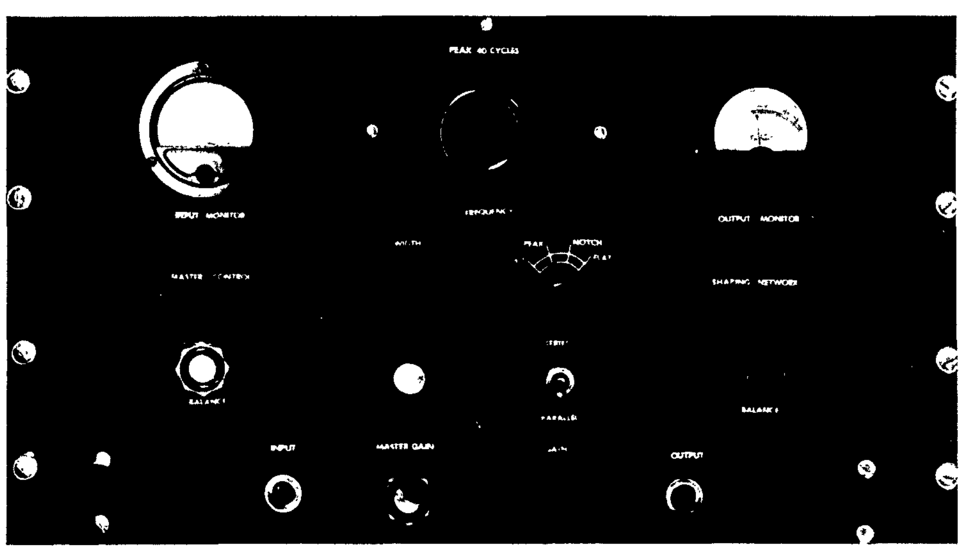

Figure 5 - Shaping network filter unit

above, we sacrificed setup time to obtain flexibility. The calibration of the system and its use will now be discussed.

USE OF THE SYSTEM

Calibration

The response spectrum must be adjusted or calibrated before each test. Since the response changes with mass on the shaker, a simulated test item must be used. This calibration is accomplished by displaying the response spectrum as seen by an accelerometer on an oscilloscope, and adjusting the networks until a desired result is achieved. An oscillator could be used for this, but we have found a commercial sonic analyzer most helpful. This device has a unit which provides a signal of constant amplitude and constantly varying frequency. The sweep rate and the high- and low-frequency limits are adjustable. The sónic analyzer also is capable of picking up a return signal and displaying it upon an oscillograph screen. Our calibration technique is to mount a dummy unit on the vibration table and to energize the shaping networks with a signal from the sonic analyzer. The operator displays the output picked up by an accelerometer on the screen of the sonic analyzer. He must then adjust the various shaping networks to produce the kind of a response characteristic required for the test specified.

Test Procedures

The complex-wave test procedure can fall in either of two general categories. First, the response spectrum can be made flat and a complex wave applied from a tape. In this case we are assuming that the signal on the tape is a good simulation of the actual environment. The tape could be made up to contain discrete frequencies superimposed upon flight records, or could contain any number of variations. The second method is to adjust the spectrum to some predetermined shape and energize the system with a noise generator. This latter technique requires more analysis to determine the shape of spectrum needed, but appears more promising since the theory is based upon an assumption of Gaussian noise.

The flat spectrum is demonstrated in Figures 6 and 7. In Figure 6, the table response, as sensed by an accelerometer, and the electrical output of the amplifier are shown before correction. The spectrum is not flat and contains a high peak at about 2,000 cps. The same two functions are shown in Figure 7 after correction. We note that amplifier output is now altered and that the response spectrum is flatter. It is interesting to note that a curve such as shown in Figure 7 would be thought of as flat by the test engineer, although it is not mathematically flat.

A shaped spectrum can be provided by the networks for testing with random noise. Here the problem is to compute the shape of spectrum desired. One way to accomplish this is to

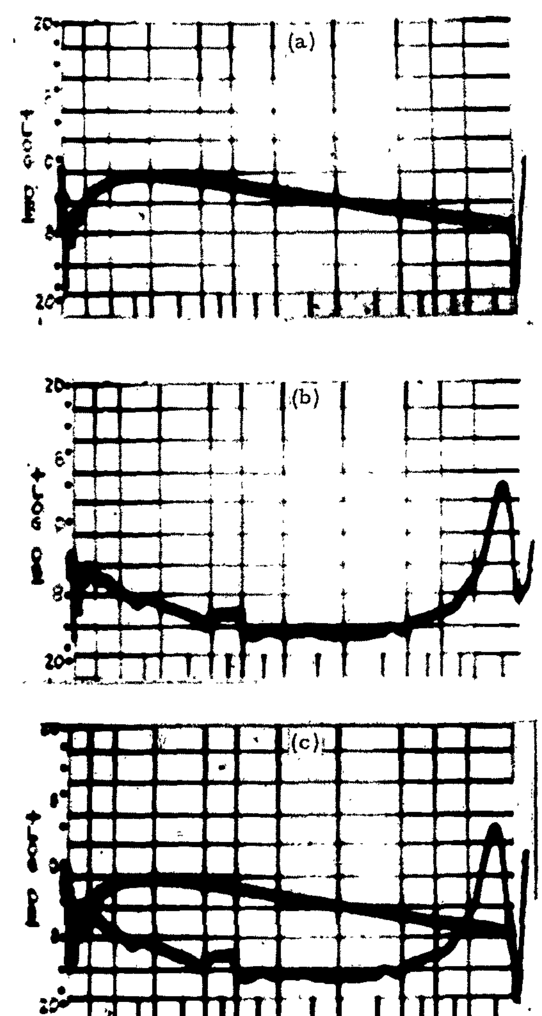

Figure 6 - (a) amplifier output; (b) table response; (c) amplifier vs table response, uncorrected spectra

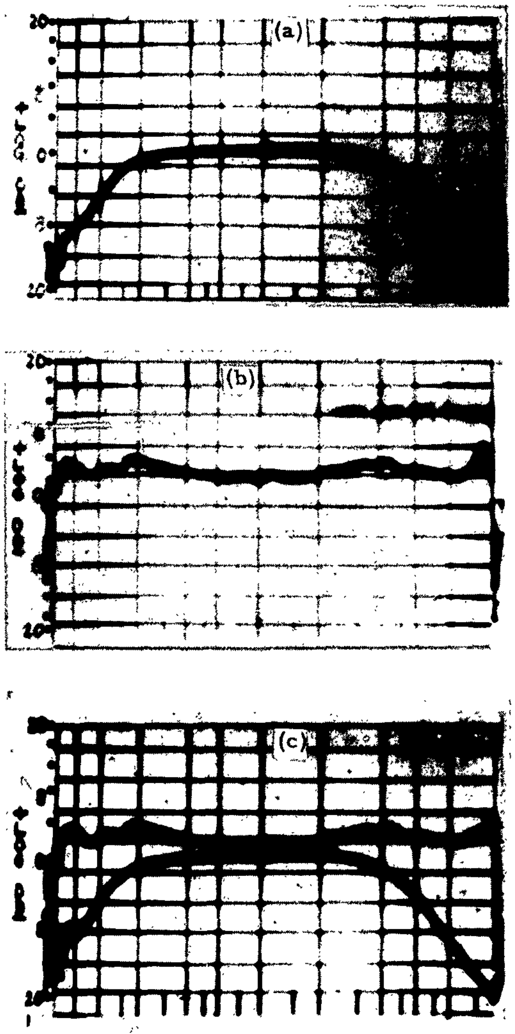

Figure 7 - (a) amplifier output; (b) table response; (c) amplifier vs table response, corrected spectra

analyze the flight tape by means of single-degree-of-freedom analogs and compute the response spectrum. This can be done if we deduce the response of a single-degree-of-freedom system as:

$$R = \sqrt{\frac{\pi}{2} f_o QG}$$

where

R = rms response of filter,

f_o = Filter resonant frequency,

Q = Magnification at resonance of single-degree-of-freedom filter,

G = Spectral density in g^2/cps.

The desired response spectrum is obtained by solving for the spectral density. A test may then be run by shaping the spectrum and using a noise generator as a signal source.

Component Testing

As we indicated earlier, we do not feel that the complex-wave method of testing is a cure-all for all of our vibration ills. We do feel, however, that it has certain particular advantages. It probably represents the closest we can come at this time to a reasonable production test and we feel therefore that any test system we devise could be used by a manufacturer of hardware for a rational production test. In addition to this, we are using our system to study the effects of complex waves on various types of components and on simple systems. We feel that we must start with a simple system at the outset rather than resort to a complicated package for our first test. Accordingly, we have used this system to test a few relays. This work is going on at the present time. We established first of all a "g"-to-failure curve for the relays. The curve plots frequency as the abscissa and the number of g's that it takes for a relay chatter to occur as the ordinate. This is somewhat of an inverse response curve. At the present time we are subjecting these relays to noise test from a noise generator. We hope then to be able to correlate the performance of the relay under complex-wave testing with its performance under sine-wave testing. The relay was chosen simply because it is to us a relatively simple mechanical component.

In addition to this we have also attempted to determine the effects of a complex-wave vibration upon some of our missile components. One of these was a stabilized platform which consists of two gyros and pick-off units. For this particular test we had measured the vibration in flight close to the mounting flange of the gyros. The particular environment here contains a very high peak at 500 cycles. We were unable to get actual flight-test tape for our test; instead we chose to shape the spectrum and use a noise generator as the excitation for the test. The flight tape was analyzed by means of single-degree-of-freedom analogs and this record plotted as spectral density vs. frequency (g^2 per cycle vs. frequency). We then adjusted the shaping networks to duplicate the spectral-density curve. Having done this we were able to energize the system with a commercial noise generator. Figure 8 shows the type of spectra that we had on this particular test. The

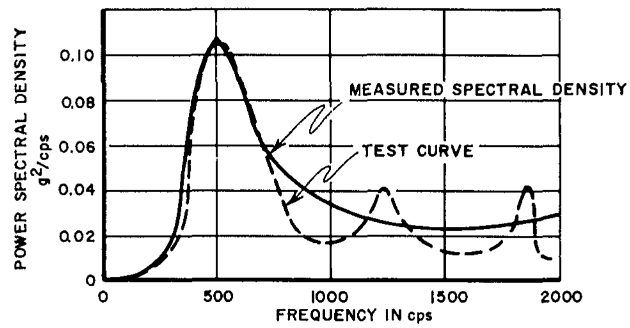

Figure 8 - Shaped spectra

reproduction is not ideal, but was deemed to be reasonable in the time allotted.

In conclusion, then, we feel we have a system, which although not ideal, does enable us to study the effects of complex-wave testing upon components and missile systems, and does enable us to come a step or two closer to the actual simulation of the environment than we have available in the customary sine-wave testing.

DISCUSSION

Krause, Jet Propulsion Lab.: I have a number of questions. The first one concerns the db accuracy you achieve?

Blevins: I believe we were within one db, plus or minus one db.

Krause: Was that a bridge T or a twin T circuit?

Blevins: We have called it a bridge T. I am not familiar enough with the different types of circuits to differentiate between the two.

Krause: I thought it was called a twin T. Passing that, have you considered the use of other types of synthesizing networks since a bridge T or twin T are not necessarily the exact type of compensation in the theoretical sense that might be required at a shake table. Have you considered a peak-notch combination or have you considered operational methods which would be the inverse of the actual model of the table?

Blevins: We have done some thinking about this. At the outset, when we began to develop these networks, we talked to a number of manufacturers, and it turned out that the time for delivery of commercial devices would be something like 18 months, though I may be wrong on that. We were able with a very small amount of effort to develop these networks in considerably less time than that, something on the order of six months, and most of that time was spent in procuring the three-ganged pot. The networks, as you undoubtedly observed, apply no phase correction at all. Our studies of the environment indicate that this is not particularly necessary, but the phase is random and if we can duplicate the spectral density curve we are getting a good simulation. At the moment there are a number of companies working on similar types of shaping networks which may be a bit more sophisticated than ours. I think our equipment has enabled us to do, up to this time, about 69 months of testing. Right at the moment, whether we will replace with something else, or not, I would not care to say. We will have to wait and see.

Krause: One more question. A large resonant load on the table usually produces a peak and a notch, and in addition there is sometimes an offset in the flat level before and after this peak and notch. Have you tried your networks on a large resonant load.

Blevins: Yes. This was exactly the difficulty we observed in the shaped spectrum that you saw on the last slide. The system being tested there actually weighed about 30 lb. We had a very heavy jig on it. so that the total mass on the table was perhaps 50 lb. This gave us a peak and a notch at one of the high frequencies, and we had a bit of difficulty in taking it out.

Kaufman, Office of the Chief of Ordnance: This particular presentation of representing a random forcing function is an excellent start and a lot more has to be done to make it automatic. For example, portraying the spectral distribution on a scope. There is another important aspect of the signature besides the spectral distribution and that is the distribution of powered spectral density or acceleration spectral density in a given band. We must not assume that it is Gaussian, because in the actual function it may be distributed quite differently. Therefore, in reducing the original forcing function to a description, we have to include this distribution in indefinite band lengths. In order to reproduce this you may eventually have to add to the network circuit not only a random generator but repeated pulse generators, sine waves, and shaking of the pulse in order to reproduce the distribution characteristic.

Blevins: Yes, I agree completely. As I indicated, we feel that we could use any type of synthesized tape we might like. Right at the moment my personal feeling is we have only a small amount of knowledge of the damaging effects of multifrequency vibration or complex vibration on components themselves. If we could establish some correlation between something we can express mathematically, Gaussian noise for example, this might be a contribution.

Cartin, Westinghouse, Baltimore: If I understood your procedure correctly, you would be required during this four- or six-hour period to maintain some low rms level on your test vehicle. Is that correct?

Blevins: Yes. I neglected to mention one thing perhaps. In calibrating our test, if we have an item we consider to be very critical, we would simply put on a dummy mass and attempt to simulate the mass of the actual item. This would depend upon the type of test that we are running and particularly the type of component we are testing. I might say that the signal we are putting in comes from a sonic analyzer and it operates at a very low rms level.

Gorton, Pratt and Whitney Aircraft: I am not an electronics man familiar with filters in enough detail to know the exact transient characteristics of your filters, but I think I can put my question in a rather general way. If we are talking of feeding random signals into this filter, this means that as you look at a given narrow band versus time, the amplitude may be changing very quickly from a very small amplitude to a very large amplitude and back again. Do you have any concern about the effect of these filters, particularly when used on their high-Q positions? Any concern about their effect on the random nature of this amplitude?

Blevins: At the present time we have not run any tests that would indicate the response of our filters to a very sharp transient. In running the shaped spectral tests which we displayed, we monitored the test body with an rms meter by putting a signal on a scope and by taking repeated cinecamera pictures of the slope of the curve. This gave us, then, a picture with a lot of hash on that had a certain envelope. The envelope thus obtained followed very closely the test curve that we showed on our slide. This would indicate that the filters are not distorting any of the transients coming from the Gaussian noise generator.

Sowell, Holloman Air Development Center: This also is for the benefit of people that aren't electronic engineers. The twin T network filter is a band rejection filter. How do you obtain your peaks?

Blevins: I will have to agree with you on the first point, that there are a number of us who aren't electrical engineers, and I am one of them. I would prefer to question Mr. King. Charlie, would you like to answer that? I point out that Mr. King did the electrical development of these filters.

King: The bridge T filter is basically a notch type of filter. In reversing the procedure and making this a peak rather than a notch, we simply use this in the feed-back loop of a vacuum tube which causes a 180 degree shift in phase.

Krause: I will direct this question to Mr. Blevins since he is a mechanical engineer. This will probably throw a monkey wrench in the works, but, above around 1,500 cps, the actual output depends on where you measure on the table and in which direction, because this becomes highly variable. As you plot across the table at, say, 15 or 20 thousand cycles per second, you get sometimes as much as two-, three- or four-to-one variations in position and direction. How do you handle this matter?

Blevins: It is, of course, very difficult to simulate an actual environment. What we have done is to take the position that we know the flight environment at some mounting point on the component, and if we are lucky we actually have this data, and we then say we are going to reproduce that environment at a similar mounting point on the shake table. Now this means,

as you point out, that other mounting points and other portions of the attached structure will see something different from what they actually see in flight. At the present time we have no way to get around this point. This is very similar to a problem which we have encountered in our sine-wave testing. There we had a large package about 30 inches long by about 20 inches across and weighing about 200 lb. We tested this package by suspending it from the ceiling, putting a driver on, and driving it. Well—we tested to a certain spec and we asked our contractors to test to the same spec, say 3 g over a certain frequency range. We always found that we both got different g levels on each edge of the package. Then we all argued at which edge it should be and how it should be done. I think this was very similar to the point that you have brought up.

* * *

ACCELERATION SPECTRAL DENSITY SPECTRUM

J. L. Wimpey, McDonnell Aircraft Corp.

This paper presents a method of applying the continuous spectrum concept to vibration analysis of electronic components. The "acceleration spectral density spectrum" concept is defined, and the mechanizing of this concept and its subsequent evaluation are discussed. Modifications of conventional vibration equipment are described and instrumentation methods for measuring the statistical variables with this technique are defined.

INTRODUCTION

In an effort to evaluate in any manner the quality level or reliability of an electronic component which is statistically representative of a production technique and tested to given environmental stresses, it is necessary as an alternative to actual flight testing of the component to provide as realistic a simulation of these environments as is possible.

With regard to this objective there is agreement throughout the industry. In going beyond this point there are many areas of controversy.

Herein is discussed a method of simulating the vibration environment found in missiles. In so doing it is the intent of this paper to present a method of applying the continuous spectrum concept to vibration analysis of electronic components.

SUMMARY

In summary this report covers the following work that has been completed to date in defining the concept, mechanizing the concept, and subsequently evaluating same. The work that has been accomplished has:

1. Tentatively defined the nature of the missile vibration environment, e.g., random amplitude variations, and white in frequency,
2. Designed a suitable voltage generator to simulate this assumed environment, and
3. Converted a conventional vibration exciter to provide a uniform acceleration response;
4. Provided networks to shape the forcing function in amplitude and/or frequency to match a known environment,
5. Designed a direct reading probability distribution meter to measure directly the statistical amplitude distribution of the forcing function,
6. Designed an acceleration detector for calibrating the acceleration at discrete frequencies throughout the bandwidth of the forcing function,
7. Established a method of evaluating the magnitude of the forcing function in terms of rms acceleration. For example, if three g's (rms) were measured at the table of the vibration exciter, this would mean that the standard deviation of the acceleration distribution would be three

g's. This would further mean that 68 percent of the time a maximum acceleration of three g's were existing at the vibration table, thus providing a yardstick by which to define desired stress levels.

WHAT IS THE ACCELERATION SPECTRAL DENSITY CONCEPT?

Evaluation of missile vibration data indicate this environment encountered is not periodic but has a random amplitude variation with a continuous distribution in frequency.

The utility of the spectrum density concept is based on the hypothesis that actual vibration environment has the following characteristics:

1. Random amplitude variation having a normal (statistical) distribution, with the mean value of the amplitude variation being at zero;
2. Frequency distribution being "white" over a given bandwidth. White frequency distribution being defined as a function whose power spectral density is constant.

In providing an analog of this environment, which will be voltage and/or current networks and transducers, certain aspects of general communications theory will be used in establishing further the nature of the problem.

In general, communications theory can be considered under two headings: 1. Classical and 2. Statistical. The first of these is quite commonplace in engineering and involves the calculation of effects for which the causes are defined functions of time, and for which the cause to affect machinery possesses negligible uncertainty. For example, the calculation of the currents which flow in a "lumped parameter" network as a result of an applied steady emf between two points of that network is a classical problem. If, however, the input to a system can be described only by a distribution function of its values or if the system itself converts causes into effects which can be described only by their statistics, then the problem is clearly one of a statistical type.

The analog of a missile vibration environment will be presented and discussed in this report. A block diagram of this analog is presented in Figure 1. If an analog had the

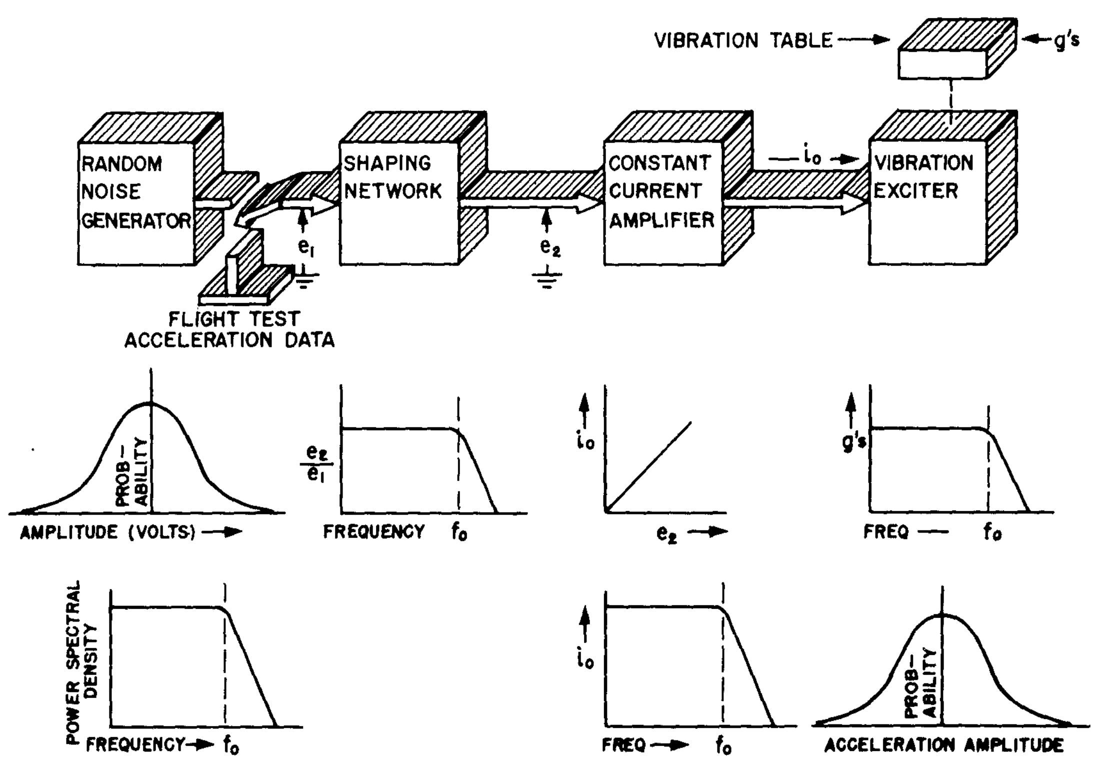

Figure 1 - Block diagram of missile vibration analog

characteristics indicated in Figure 1, it can be assumed that the instantaneous acceleration g in the frequency band f_1 to f_2 will have a normal probability distribution about zero as a mean, and given by the formula

$$P(a)da = \frac{1}{\sigma\sqrt{2\pi}} \Sigma^{\frac{-a^2}{2\sigma^2}} da \ . \qquad (1)$$

Where $\underline{a}$ is a given instantaneous acceleration, $P(a)da$ is the probability that $\underline{a}$ lies in the interval between $\underline{a}$ and $\underline{a} + da$, and σ is by definition the rms acceleration and is known as the standard deviation.

If an accelerometer, with an acceleration response that is flat up to f_o, were placed on the shaker, the accelerometer voltage will have the same amplitude distribution as that of the output voltage distribution of the noise generator. The arrangement of Figure 1, thus seems to fulfill the requirements of providing an analog of the assumed missile environment.

Upon one's first exposure to this concept of vibration testing a logical question arises with regard to what single number can be used to express the vibration in a specified band or in the entire spectrum. In arriving at such a number it appears that whatever this number is, it should be possible to obtain with present day, commercially available instrumentation. Assume for now that this be the case. Consider next, any arbitrary voltage waveform vs. time that is corresponding to acceleration, and explore the methods of measurement. Measurement of the peak acceleration does not yield much because voltmeters designed to read peak for a sine wave do not necessarily read peaks for a complex wave. The average of the absolute magnitudes would be more suitable, such as would be obtained by full wave rectification, but this does not lend itself well to statistical treatment.

The root mean square (rms) of the instantaneous acceleration appears to be the best choice, because it is by definition the standard deviation of the instantaneous accelerations about zero as a mean, and thus lends itself to statistical analysis. Also, the rms value is the old measure of ac voltage in electronic circuits.

A METHOD OF MECHANIZING FIGURE 1

There are commercially available random noise generators (General Radio random noise generator No. 1390A) that are known to have a normal amplitude distribution and are "white" in frequency from a few cycles to hundreds of kilocycles. The type of generator used to date at MAC in carrying out these investigations, employs a thyratron in a magnetic field. The frequency spectrum of a gas-tube type of generator is known to have the aforementioned frequency characteristics.

Modification of Vibration Exciter

To facilitate spectrum density vibration analysis, it is necessary that the vibration exciter provide a uniform acceleration over a wide range of frequencies. The following tests were conducted to determine the possibility of adapting presently available equipment to this purpose.

An MB vibration exciter (model C-11) was used for this investigation which consists of a rack-mounted audio oscillator, electronic amplifier and power supply, field supply, and vibration exciter. The vibration exciter is made up of the following: main field coil, signal generator field coil, driver coil, and signal generator coil. A current of one ampere in the driver coil produces a force of 10 lb at the table. Movement of the signal generator coil in its field produces a voltage that can be used to determine acceleration and displacement in the following way:

$$d = 100/f \ E_{mv}/Sen_{mv} \ 0.001, \qquad (2)$$

$$g = 0.000512 \ f \ E_{mv}/Sen_{mv} \qquad (3)$$

where

d = double amplitude inches,

f = frequency cycles,

E_{mv} = Output of signal coil in millivolts shunted by 20,000 ohms,

Sen_{mv} = Sensitivity millivolts/0.001 inch displacement,

g = Acceleration gravity units.

The vibration exciter, as normally used with full voltage feedback has a current response as shown in Figure 2. This is obviously an undesirable response with a continuous spectrum applied.

Forces are produced between the field structure and the armature coil suspended in the

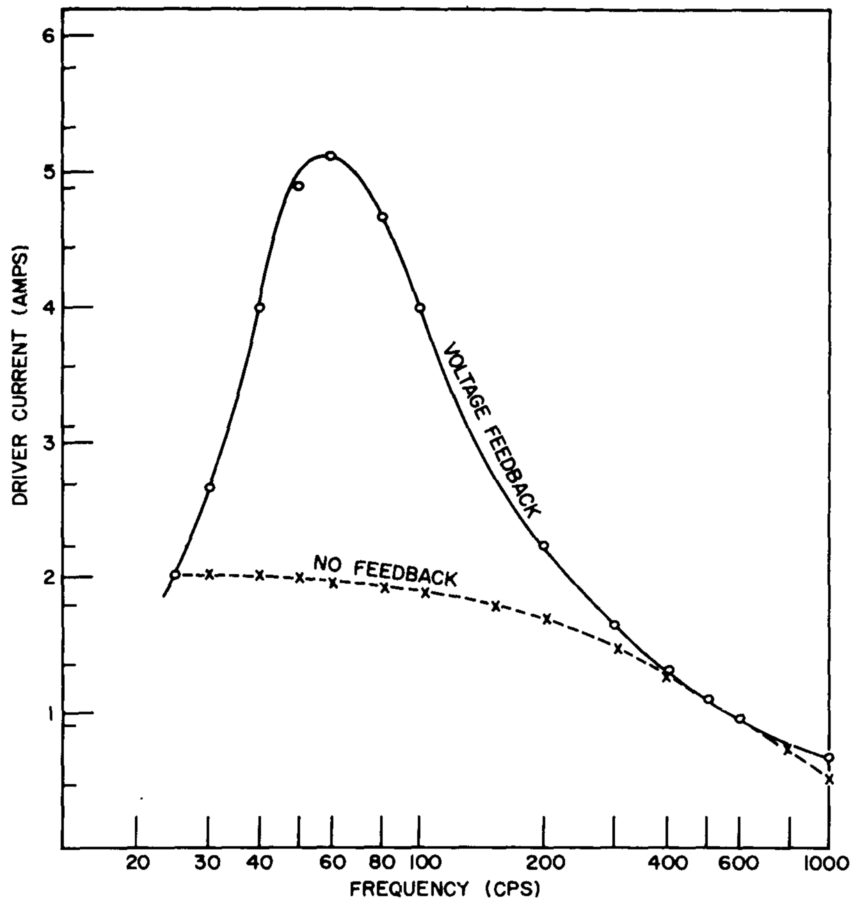

Figure 2 - Driver current vs. frequency

magnetic field. The force on the armature is the sum of the forces acting on each coil conductor in the magnetic field. When converted to pound-inch-second units, the basic force relationship is:

$$F = 0.885\ 10^{-7}\ BLNI \tag{4}$$

where

F = force, pounds,

B = flux density, lines/in^2,

L = length of single armature turn, inches,

N = number of turns in flux gap,

I = conductor current, amps.

For any particular shaker design, L and N are fixed properties of the armature, and Equation (4), indicates that the force produced by such an armature is proportional to the current flowing through its conductors if the field strength is constant.

To apply the continuous spectrum concept to this type of vibration equipment, it becomes necessary to alter the performance of the power amplifier driving the vibration exciter in such a manner as to provide a constant driving current independent of load instead of a driving voltage independent of load. This is to say that the output of the amplifier should be a constant current source instead of a constant voltage source.

If the driving current is constant with respect to frequency, it follows that the acceleration at the table will also be constant for a given mass that is on the table.

If the networks between the random noise source and the table of the vibration exciter are linear, the acceleration distribution at the table

will be the same as that of the voltage (amplitude) distribution coming from the noise generator.

The first step in providing a constant current source to drive the vibration exciter was by way of eliminating the negative voltage feedback in this particular amplifier. By this method, alone a much improved current response, Figure 2 (no feedback), was obtained. Current feedback gave the desired effect. The best system of current feedback found that does not require a circuit change within the power amplifier is given in Figure 3. The current response of the amplifier with the current feedback loop is shown in Figure 4.

It was determined that 1,000 cycles is essentially the upper limit of the system so a low-pass network is employed between the noise generator and the converted current amplifier with a cut-off frequency of 1,000 cps.

Thus, the basic blocks of Figure 1 have been implemented, calibrated, and found to yield characteristics sufficiently close to those required.

INSTRUMENTATION REQUIRED THAT IS NOT COMMON TO DISCRETE FREQUENCY TESTING

Inasmuch as the concept herein discussed is statistical in nature, the important item to be concerned with is determining the type of distribution the analog forcing function may have. A direct reading probability distribution meter has been designed to determine the probability distribution of stationary random and non-random functions. Experimental tests have been made using Gaussian random noise with a resulting accuracy comparable to the average requirement for an electronic measuring instrument.

Acceleration Detector

The acceleration detector consists of a tuned reed sensing device which frequency modulates a 4.5 megacycle oscillator. This frequency modulated signal is then amplified and fed into a ratio detector. A block diagram is provided in Figure 5. The sensing device is made up of five reeds tuned to frequencies of 157, 413, 541, 812, and 981 cycles/second. The reeds are mounted on a common plate in such a way that each becomes a small capacitor which is used to tune the oscillator tank circuit. When the assembly is subjected to a vibration environment, the reeds are excited by frequency components of their natural frequency with a reed displacement proportional to the exciting acceleration. The reed displacement changes the capacitance between the reed and the plate, therefore changing the oscillator frequency.

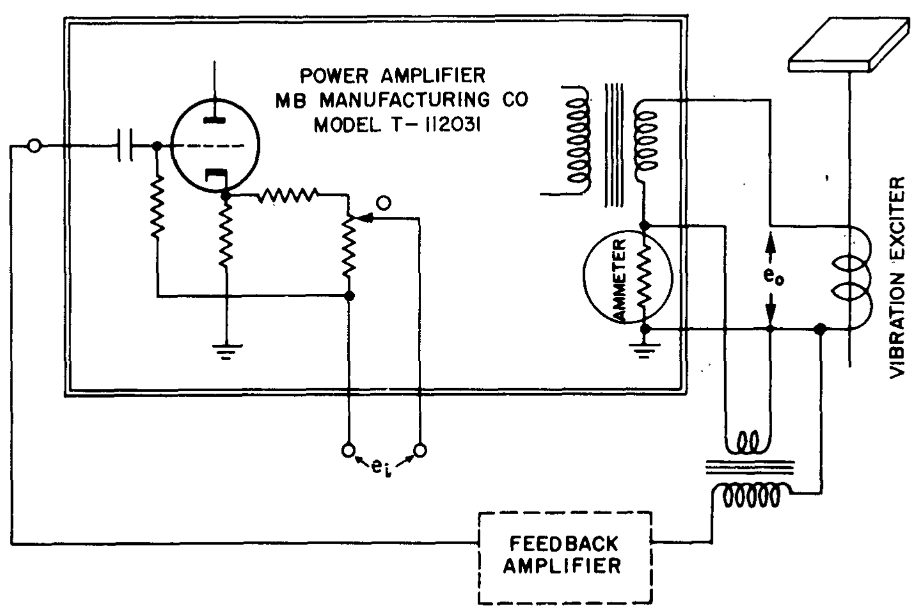

Figure 3 - Current feedback method employed

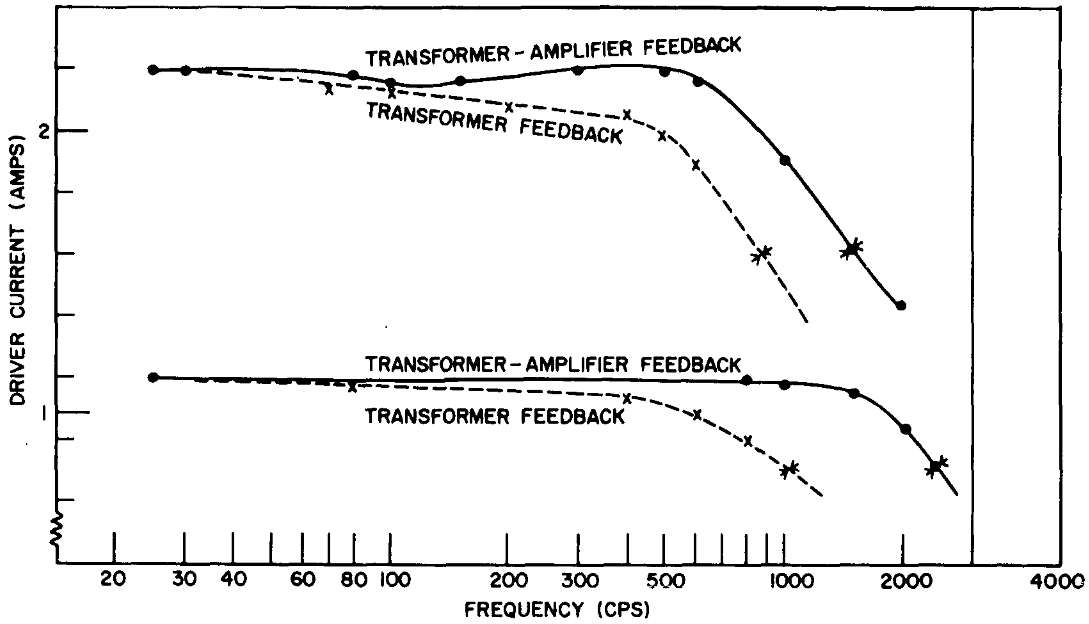

Figure 4 - Current response of amplifier for two different levels of input voltage with a transformer as feedback and a feedback loop with transformer plus voltage amplifier

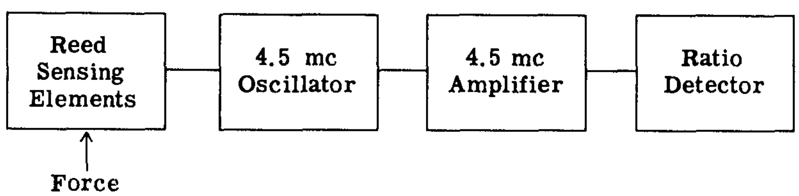

Figure 5 - Block diagram at acceleration detector

A switching arrangement is provided at the reed assembly to monitor the amplitudes of each reed. The reeds are calibrated at discrete frequencies so that when a complex wave is applied, acceleration can be determined at specific frequencies throughout the complex wave bandwidth. Figure 6 is a schematic diagram of the complete acceleration detector.

True Reading rms Meter

In order to read the rms acceleration at the shake table it is desirable to have a true reading rms meter. There is commercially available such an instrument (Model 320 Ballantine) but it appears that an average reading meter can be used with an appropriate scale factor to derive the rectified dc voltage V_o from noise:

$$V_o = \int_{-\infty}^{\infty} VP(V)dv = 2\int_{o}^{\infty} VP(V)dv \qquad (5)$$

where the second integral follows from the first if $P(V)$ is symmetrical about zero.

If the noise is Gaussian, the probability $P(V)dv$ that the instantaneous voltage lies between V and $V + dv$ is of the same form as Equation (1). Here we are considering a voltage distribution instead of an acceleration distribution because ultimately we will measure a voltage with a known functional relationship to acceleration. Equation (1), written in terms of voltage becomes

$$P(V)dv = \frac{1}{\sigma\sqrt{2\pi}} C^{\frac{-V^2}{2\sigma^2}} dv .$$

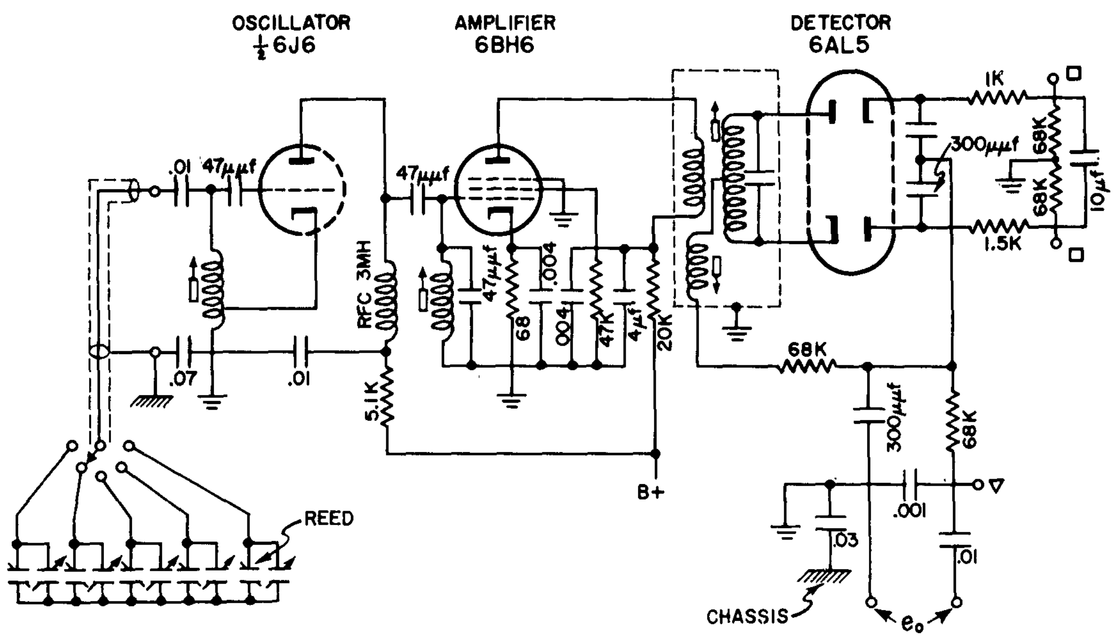

Figure 6 - Schematic of acceleration detector

Substituting this into the expression for V_o yields

$$V_o = \frac{2}{\sigma\sqrt{2\pi}} \int_0^\infty V \cdot e^{\frac{-V^2}{2\sigma^2}} \, dv . \tag{7}$$

The solution of Equation (7) yields the following relationship of V_o to σ. Equation (7) can be written in the following standard form of definite integrals,

$$\int_0^\infty x e^{-x^2} \, dx = \frac{1}{2} , \tag{8}$$

by letting $\qquad x^2 = \frac{V^2}{2\sigma^2}$

and thus, $\qquad x = \frac{V}{\sqrt{2}\sigma} ,$

$$dx = \frac{dv}{\sqrt{2}\sigma} .$$

By making the proper substitution of variables we have the following:

$$V_o = \frac{2\sigma}{\sqrt{2\pi}} . \tag{9}$$

Since the meter is calibrated to read the rms value of a sine wave, the indicated voltage V_i = 1.11 (i.e., $(\pi/2\sqrt{2})$ times V_o, where V_o is the average or rectified value). Thus,

$$\frac{V_i}{\sigma} \quad \frac{\sqrt{\pi}}{2} = 0.886 \text{ or } -1.05 \text{ db.}$$

Average reading meters thus read 1 db low on Gaussian noise provided that no overload occurs on the peaks.

EVALUATION OF SYSTEM

If we wished to redefine the nature of the missile environment, and include a periodic vibration contained in the random vibration, a signal proportional to acceleration could be mixed in prior to the constant current source that is driving the vibration exciter. This periodic vibration may be, for example, the fundamental and/or harmonics of a ram-jet engine.

This system can also be used for reproducing the actual acceleration environment that may be available from in-flight recording which can be fed into the system where the noise generator is used, as indicated in Figure 1. Selection of shaping networks can be made by switching arrangements that will modify the amplitude

and/or frequency characteristics of any forcing function.

Correlating the effect of any complex forcing function to a discrete frequency-forcing function is rather difficult, inasmuch as the time honored yardstick "severity" has little meaning with regard to these two approaches. What may be severe for one structure may not be so for another structure. It appears that the basic question one has to ask is—which approach is a more realistic simulation of the actual environment?

It appears at this point that with the techniques developed herein, considerably more realism can be introduced by continuous spectra testing of electronic components than by discrete frequency methods. It is anticipated that some confirmation of this can be made by further evaluation of this concept.

With regard to testing a complete electronic package, it appears that methods discussed herein would be most useful in proving a design. It may have less merit in development testing because in this case it is desirous to know specific resonances in the structure which can be found with discrete frequency testing, and thereby provide design requirements for vibration isolators.

There has been an attempt made in this report to show how to mechanize a philosophy that says that missile vibration environment is entirely random. It has been concluded by others in the field that a good example of this is vibration resulting from aerodynamic turbulence.

Analysis made by people at JPL indicate that with most rocket motors, however, some discrete frequencies also appear. Most liquid rocket motors give a single frequency (that is, periodic), where solid rocket motors produce a display of several discrete frequencies. With this being the case, the purely random vibration amplitude can be altered by mixing simulated periodicity into the system. The degree of periodicity that one may wish to simulate could be measured in terms of the auto-correlation function of the composite vibration analog. Inasmuch as analog computing techniques will provide the auto-correlation function, this appears to be a method of defining the nature of a specific missile vibration environment that would lend itself well to standardization.

DISCUSSION

R. Krause, Jet Propulsion Lab.: I would like to agree with Mr. Wimpey that where you have low loads on the table, a constant current will effectively give you a constant acceleration. However, we should be aware, where we have resonant loads, that reaction back on the table occurs and the constant current will no longer give us constant acceleration.

A rough rule of thum would go like this. Supposing we have a spring and a mass attached to the table with a Q of, say, ten. Then, if the mass ratio is less than one-tenth, the approximation of constant current is good. If the mass ratio was greater than one-tenth, we have to take other means to get a flat response.

Wimpey: Yes, sir. I agree fully, and this is usually the case when we are talking about tables that can handle the testing of vacuum tubes and small components where the ratio is so small.

Bubb, Phillips Petroleum: I used it in the Air Force and tended to be pestered a great deal by noise problems of all sorts. I hadn't really realized that there are mechanical noise problems. I was interested in Mr. Kearn's remarks concerning more time variant intervals, and for those of you who want to look at these problems—the statistical results of vibration—you will find some 750 articles upon which I had collected data. They are now, I believe, on deposit at the Aeronautical Research Laboratory at Wright Field.

General Electric has a simplified version of the "yellow peril," and I am told it is more comprehensible. I believe the gentlemen here from GE can tell you the title. Bill Jordan told me it has been published as a government publication, and I believe it is in current use in connection with communications problems.

There is also another treatment of this, entirely in the time domain, where you are not concerned with any linear integral transformation problems, such as the frequency type of analysis. I'd be very happy to pass on any information I have about these other applications and, in particular, to those of you who may be interested in understanding this method of attack. I am sure Mr. Kearns, for one, would be interested.

CONFIDENTIAL

C. B. Cunningham, NRL: Mr. Wimpey, I have two questions: first, what weight objects were you testing? That is, the general range of objects; second, how much power were you using in this table? I'd like to have a little more description of your power amplifier.

Wimpey: To answer your first question, the weights of packages were in the order of half a pound or a pound, containing several vacuum tubes and some resistors and capacitors. These were the biggest loads we handled.

Now with regard to your second one, we had a hundred-watt amplifier, and at a single frequency the shaker is capable of delivering something around 7 g to an object of this weight. When you spread this power over a thousand cycles of bandwidth, the maximum capability of this shaker is 0.25 g rms and it appears that 0.25 g rms is a far more severe environment than you might imagine. Vacuum tubes have an awfully hard time living in this environment.

Cunningham: Just as a comment, if a vacuum tube has a hard time living at 0.25 g, we are in real serious trouble.

Wimpey: Well you have 0.25 g rms over this entire range of 1,000 cycles. That is equivalent to 7 g for every cycle over the same range for sine wave testing and if you were to excite the resonances of any structure with a high Q such as a vacuum tube, 7 g will probably cause you trouble.

Cunningham: In Project Vanguard we are presently testing to 20 g rms, from 10 to 2,000 cycles, and this means that we must have a great deal more acceleration than you are talking about. That is why I was interested.

Wimpey: Is this acceleration environment constant?

Cunningham: We are talking about white noise, 20 g rms.

Wimpey: You have got problems. (Laughter)

Granick, WADC: I have been trying for sometime to find out more about this white noise; where it comes from, and how is it possible to transmit it through structures with discrete resonances.

Wimpey: If you provide an environment that has a flat acceleration response and you are getting, at that table, a distribution of amplitudes that is normal, and you attach to the table a structure that you are concerned with, then you can measure at any point the effect on this structure of this environment. Resonances will appear and you will get an amplification of these resonances which will occur in a very random manner. The resonances—the output so to speak—of this system, with a normally distributed noise input, will give you something that looks like a sound wave that is jumping about randomly, and in turn, every time it gets excited you will get amplification at the other end of the structure, the same as you would if you were vibrating it at a discrete frequency.

Granick: I believe my question was misunderstood. This may be beyond the scope of your paper. However, the question I was interested in was the actual service environment; where does this white noise come from? How does it get on the structure? And how can it be transmitted through a structure that has discrete resonant frequencies?

Wimpey: I can say that one way that environment can be created in an actual field application—in a missile, for instance—would be aerodynamic buffeting. That is a big order of random vibration that is transmitted throughout the structure.

Cunningham: I think we should make a distinction between white noise and random noise of vibration because it is not necessary for it to have a flat spectral distribution or spectral density to be random, or that it have a Gaussian distribution of acceleration. In fact, most of the vibration that we see in the actual missile environment, for example, will be random. The way it arises is from any process that doesn't have something repetitive about it, such as, for example, turbulence, noise in the stream of a jet; any situation where there isn't a feedback mechanism, for example, which is built on a sinusoidal steady level.

Now, in the laboratory, we may test with white noise. In other words, we may test with a spectrum that is flat, because we will fill in the valleys in many cases and assume that resonant peaks could occur on a different flight in the frequency locations that are valleys in one flight. So we have white noise to test with in the laboratories as a specification. But it does not ordinarily occur in the missile environment. Random vibration does.

* * *

PART III

SHOCK AND VIBRATION TEST FACILITIES AND METHODS

DEVELOPMENT OF A HEAVY-WEIGHT (2 TO 20 TONS) SHOCK TESTING MACHINE

R. Gareau, BuShips

High-impact shock testing machines available today cannot adequately test equipment weighing more than 4,500 lb. This paper describes underlying considerations in the design of a heavy-weight (2 to 20 ton) high-impact shock machine. The present status of the project is described and a brief description of the proposed machine is given.

INTRODUCTION

This paper will present the progress that has been made in developing the design of a so-called "heavy-weight shock machine." This machine, if it is ever built, will be the largest shock testing machine in the world. It will be able to shock test equipment weighing up to twenty tons. It will mean that most auxiliary shipborne equipment, including main propulsion diesels, will be capable of being shock tested.

But before getting involved in the specific details of this machine, some background data will be given relative to the machines that have been or are now being used. In addition, the basis for the design of the heavy-weight machine will be presented.

BACKGROUND

Until the beginning of World War II, shipborne equipment was shock tested on the 250 ft-lb machine. This machine derived its name from the fact that it consisted of a swinging hammer which weighed either 50 or 83-1/3 lb and could fall a maximum height of 5 or 3 ft. The hammer struck a steel angle bolted to the top of a wooden test bulkhead. The equipment under test was bolted to this wooden bulkhead. This machine simulated the shock motions that ship's structure experienced when exposed directly to gun blast resulting from the firing of the ship's own guns.

The shock severities on this machine dropped off very quickly when equipment weighing more than 50 lb was mounted to the test bulkhead. The maximum shock severities for small equipment weights are about 10 ft/sec at a frequency of 50 cps. Additional details of this machine are included in Reference 1.

At the beginning of World War II, the British were the first to experience the very extensive damage that could be caused by the detonations of explosives close to the ship's hull. These detonations would cause little or no damage to the hull structure but would seriously damage all kinds of ship's equipment.

The British realized that much of their equipment had to be ruggedized quickly. Without paying too much attention to a realistic simulation of field conditions, they put together a shock machine which duplicated the kind of damage that was experienced under combat conditions.

This machine also used dropping hammers which struck a steel anvil plate and thus produced

the desired shock motions. The machine was provided with two separate hammers each weighing 400 lb. They were so arranged that they could be dropped a maximum five ft and impart shock motions in three mutually perpendicular directions.

This machine was soon adopted by the U. S. Navy and its use spread from naval laboratories to a large number of commercial laboratories. It is known in this country as the "Class HI Shock Machine for Light-Weight Equipment." Its use is specified in Specification MIL-S-901B [Reference 2] and at least 50 of these machines have been built and are now being used. They have proven very useful in ruggedizing a large number of items of equipment weighing up to 400 lb. But as with the "250 ft-lb machine" the shock severities start to drop off quickly when the weight of the tested equipment is greater than 250 lb. For that reason, the upper weight limit has been established at 250 lb.

For a weight range of a few pounds to 250 lb, the maximum peak velocities vary from 18 ft/sec to 12 ft/sec and these velocities are associated with frequencies of about 100 cps. Additional details of the shock characteristics of this machine are included in Reference 3.

Since the effects of near-misses were felt by all kinds of equipment, steps were taken as early as 1943 to build a machine capable of shock testing equipment weighing more than 250 lb.

Again, influenced by the fact that we were engaged in a World War, not too much effort was made to simulate actual shipboard shock motions. As a matter of fact, not too much information was available at that time on the nature and magnitude of shipboard shock motions.

So, another swinging hammer machine was quickly designed and built. Provision was made this time for a 3,000-lb hammer swinging in an arc and capable of dropping a maximum of 5.5 ft and hitting the underside of a steel anvil plate.

Various tests and calibrations established the weight limit of this machine at 4,500 lb. The machine has become known as the "Class HI Shock Machine for Medium-Weight Equipment." Details of the test procedures that are employed are included in Reference 2.

Maximum shock velocities of 10 to 12 ft/sec at a frequency of 70 cps are characteristic of this machine. Equipment weighing up to 9,000 lb has been tested, but at this weight range the peak velocities drop down to 5 or 6 ft/sec and it is therefore considered to be a substandard test. Additional details are included in References 4 and 5.

Some consideration was given back in 1944 to building a larger shock machine with a hammer weighing 16,000 lb. This proposal was never seriously considered beyond the very preliminary design stages because of the difficulties that could be foreseen. One stumbling block was the problem of stress concentration at the point of hammer contact. Another reason for postponing this development was the desire to simulate more closely actual shipboard shock motions.

By 1952, sufficient data had been obtained on which to base a study to determine the nature and magnitude of representative shipboard shock motions. Some of this information included data gathered by the British and exchanged with us under the terms of an Information Exchange Project.

Both the Naval Research Laboratory and the David Taylor Model Basin made an analysis of the available data and made recommendations on the type of shock motions that can be expected aboard naval vessels. These recommendations were used as a basis for specifying typical shock motions that a heavy-weight shock machine should simulate.

BASIC DESIGN CONSIDERATIONS

It was decided at the beginning of this development that the desired shock simulation could be more conveniently expressed in terms of velocity-time curves. The principal reason for this choice was that the most reliable full-scale data was obtained by using velocity meters.

It should be realized that the type of shock motions that actually exist aboard ship are extremely complicated and that for practical reasons some compromise had to be made as to the kind of shock motions that could be practically simulated.

Taking all these factors into consideration, the best compromise that could be arrived at is shown in Figure 1. The required shock motions have been expressed in terms of velocity-time curves for different equipment weights ranging from two to twenty tons.

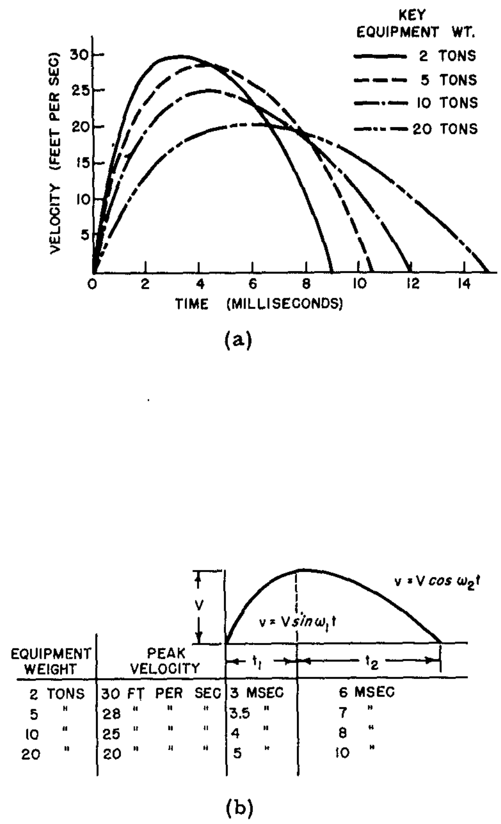

Figure 1 - (a) maximum severity-velocity time curves, (b) tabulated data

The curve is not symmetrical and the initial motion may be considered to be principally governed by the character of the pressure-time curve of a particular underwater detonation. Subsequent motions are slower and governed principally by the spring constants of the structure supporting the item of equipment.

Peak velocities of 30 ft/sec are associated with equipment weights of two tons and peak velocities of 20 ft/sec are associated with equipment weights of twenty tons. Peak accelerations and displacements derived from these velocity-time curves are shown in Table 1.

It should be realized that there is no direct comparison between these derived values of accelerations and the shock design accelerations that are specified in general specifications for the design of shock-resistant equipment.

Once the desired shock motions had been established it was not difficult to determine what additional criteria should be considered

TABLE 1
Tabulated Design Parameters

Equipment Weight	Peak Vel. (ft/sec)	a_o (g's)	a_2 (-g's)	S_1 (in.)	S_2 (in.)
2 Tons	30	488	244	0.688	1.373
t_1 = 3 msec	15	243	122	0.344	0.686
t_2 = 6 msec	8	130	65	0.184	0.367
5 Tons	28	390	195	0.750	1.50
t_1 = 3.5 msec	14	192	96	0.374	0.750
t_2 = 7 msec	8	111	55.6	0.214	0.428
10 Tons	25	305	152	0.764	1.525
t_1 = 4 msec	12	146	73	0.367	0.734
t_2 = 8 msec	8	97.7	48.7	0.245	0.489
20 Tons	20	195	97.5	0.763	1.53
t_1 = 5 msec	10	97.5	48.7	0.382	0.765
t_2 = 10 msec	8	78	39	0.306	0.612

in the development of this machine. The design criteria included:

1. A shock test table, 8 ft by 16 ft and provided with suitable hold-down arrangement so that equipment could be mounted and operated during the shock test.
2. The overhead and area surrounding the test table should be kept clear to facilitate hoisting and handling the experiment during the test set-up.
3. Equipment under test should be subjected to horizontal as well as vertical shock without the necessity of remounting.
4. The shock machine should be adjustable and have a flexible control system in order to allow a wide variation in the magnitude of the shock motions.
5. The shock-test table should be rigid enough so as not to distort the desired shock motion.

A listing of all these requirements represents a formidable array of conditions. There was some doubt at the beginning of this project if any reasonable progress could be made. For that reason the project was split into four phases and could be conveniently closed out when it was obvious that no progress could be made at any particular phase. These phases were:

Phase 1. Feasibility study
Phase 2. Preliminary or contract drawings
Phase 3. Production and detail drawings

Then the construction of the prototype follows.

GENERAL DETAILS

A design contract was granted in January 1955 to Power Generators, Inc. to undertake Phases 1 and 2 of this development. Phases 1 and 2 have been completed and final reports have been submitted, References 6 and 7.

As part of Phase 1 various schemes were considered. These included: dropping test table, multiple swinging hammer, air actuated hammer, torsion springs, linear spring using steel in bending, rubber spring, electro-magnetic devices, explosives, and liquid springs.

The scheme that was finally adopted involves the use of liquid springs and employs the compressibility of oil to store the desired amount of energy. These springs can theoretically develop the required forces and at the same time furnish the desired amount of control.

Some idea of the maximum forces and pressures that are required are given in Table 2. The peak force of 36 x 10^6 lb is associated with the two-ton test load.

It has proven comparatively simple to design a mechanism capable of producing these large forces. The problem, however, of developing a mechanism capable of releasing these forces quickly enough has been the most difficult. This problem has been solved by the ingenious application of a wedge-release mechanism.

DESCRIPTION OF HEAVY WEIGHT SHOCK MACHINE

The machine consists of a very rigid welded steel table, 8 ft wide, 16 ft long and about 8 ft deep. The table is connected to a dual liquid spring assembly which is used to apply the required accelerating or decelerating forces. Two of these spring assemblies provide for horizontal or vertical shock motions. Load cells are incorporated in order to assure equal weight distribution prior to the shock test. The general arrangement is shown in Figure 2.

The liquid spring, Figure 3, used in this machine is a device which generates a linear force-displacement characteristic in the same manner as a steel spring. The liquid spring uses the compressibility of oil as the spring medium. For this application, a proprietary product, Comproil 118, is employed. The proposed arrangement consists essentially of two springs within one cylinder. The lower chamber is the accelerating spring and the upper chamber is the decelerating spring.

The volume of the liquid spring chambers is adjusted to provide the desired spring rate. The volume variation is accomplished by adding or removing mercury from the auxiliary chambers connected to the main upper and lower chambers.

The desired force is obtained by compressing the oil on the lower or acceleration side of the

TABLE 2
Derived Design Data

Test Weight (Tons)	Initial Accel-a_o (g)	Final Accel-a_2 (-g)	Initial Force-F_o (x 10^6 lbs)	Final Force-F_2 (-x 10^6 lbs)	Initial Pressure-P_o (psi)	Final Pressure-P_2 (psi)
2	488	244	35.1	15.1	17,800	12,500
	243	122	17.5	7.56	8,860	6,250
	130	65	9.34	4.03	4,730	3,330
5	390	195	30.4	13.25	15,400	10,950
	192	96	14.95	6.53	7,570	5,400
	111	55.6	8.65	3.78	4,380	3,110
10	305	152	26.8	11.85	13,600	9,800
	146	73	12.8	5.7	6,500	4,700
	97.7	48.7	8.58	3.8	4,350	3,600
20	195	97.5	21.0	9.55	10,650	7,900
	97.5	48.7	10.5	4.76	5,330	3,940
	78	39	8.4	3.82	4,250	3,160

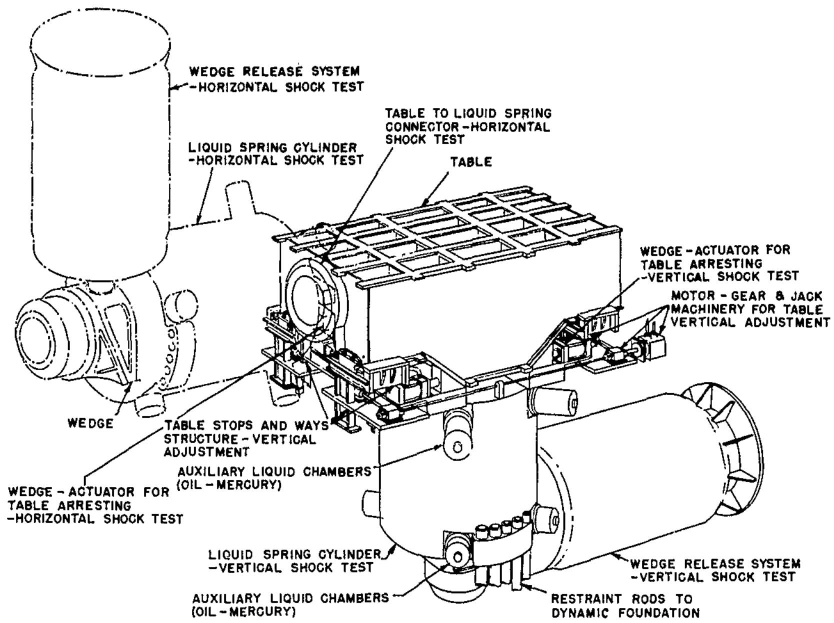

Figure 2 - General arrangement of heavy-weight shock machine

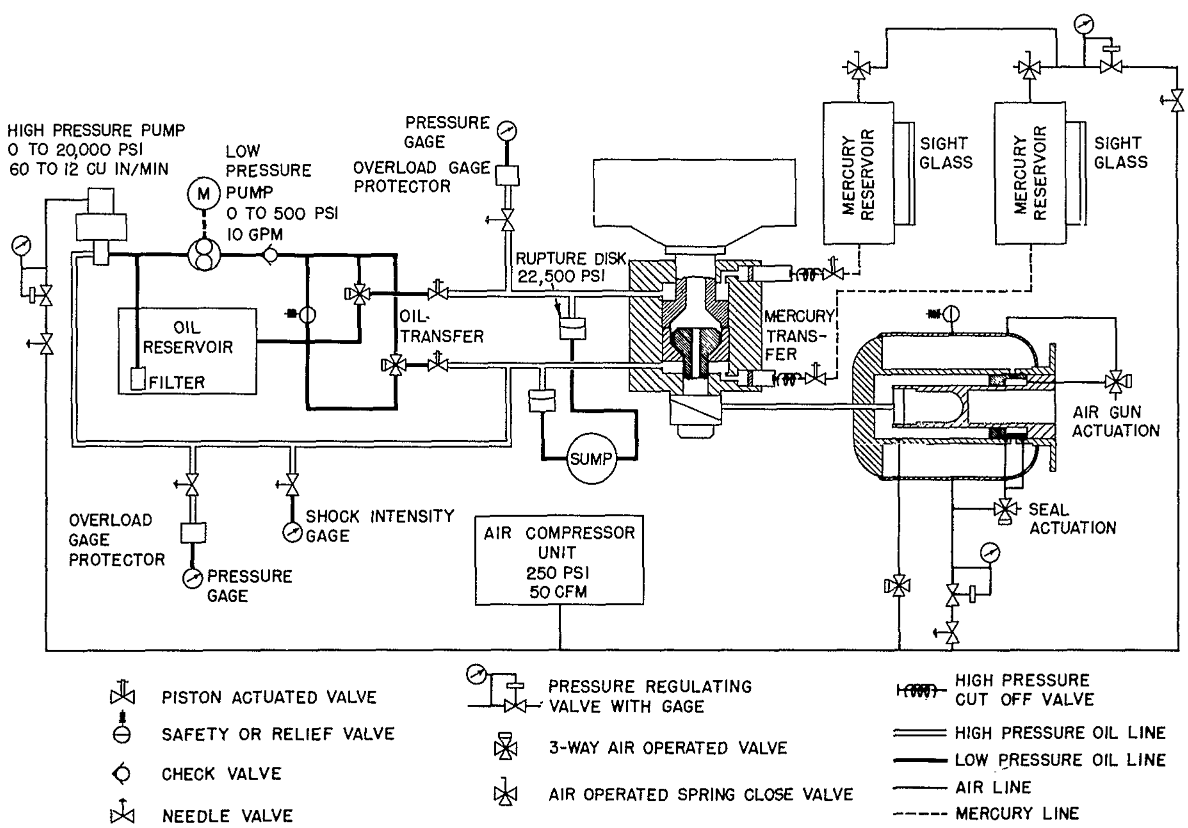

Figure 3 - Schematic of liquid spring

piston. The piston is restrained from moving by means of a tensile rod concentric with and bearing on the inside surface of the piston. The opposite end of the tensile rod is fitted with a wedged surface which bears against a similar wedged surface and thus prevents motion of the piston until the mating wedged surface is forcibly removed. Theoretically an angle can be chosen for the wedge such that it is at the point impending self-locking. Under this condition, the wedge can be moved by a small force. If the wedge is moved quickly enough, then no restraint is placed on the shock test table during its accelerating phase. This principle is used in the proposed wedge release mechanism.

The mechanism required to remove the wedge quickly has turned out to be more complicated than the actual shock machine. The wedge release mechanism consists of three concentric cylinders. The outside cylinder serves as an air reservoir for accelerating the piston which slides in the midcylinder. The innermost cylinder is fitted with a second piston which is connected to the wedge that must be quickly removed. The innermost cylinder also serves as a liquid buffer and a means of positioning the wedge after a test. Water is included between the two pistons. This water serves as a medium to transfer the kinetic energy of the piston being accelerated to the pressures needed to actuate the wedge.

Operationally, air is applied to the piston in the midcylinder. This piston moves and exposes the exahust ports of the outer cylinder and thus applies the full air pressure and accelerates the midpiston to the required velocity. The water that has been introduced between the two pistons is forced out until, near the end of its stroke, the midpiston now moving with maximum velocity closes the exhaust ports and begins to compress the water remaining at the extreme end of the midcylinder. This increase in pressure actuates the wedge release and the required test motions are applied to the equipment.

It is realized, of course, that the studies to date have been largely theoretical. There are still a number of problems to be worked out. In the design of such a massive machine, not all design problems can be solved by analysis or preliminary layout. For these reasons, it is foreseen that experimental testing of some of the more critical items will have to parallel the development of the production drawings.

CONCLUSIONS

In conclusion, studies so far have shown that it is feasible to construct a heavy-weight machine that will be capable of simulating the required shock motions that shipboard equipment in the weight range of 2 to 20 tons can be expected to encounter. These shock motions correspond to 30 ft/sec for the 2-ton weight and 20 ft/sec for the 20-ton weight. Frequencies of 84 cps and 50 cps are associated with these velocities.

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DISCUSSION

Muller, Sperry Gyroscope: I would like to ask Mr. Gareau what displacement is associated with those velocities.

Gareau: Well, the displacement varies, of course, but it is not greater than 2 inches.

Muller: If it is 2 inches, how do you get the table back again to a stalled position? How do you bring it to a stop? You show a velocity time curve there.

Gareau: Well, I failed to mention at the time one other condition we set up for tests on this machine—the subsequent shock motion should not be greater than the initial half cycle pulse. In the particular diagram which you saw, provision was made to incorporate on the four corners of the test table, a wedge device which would move up with the table. Therefore, it is conceivable that you would not have any subsequent motion, that once the half cycle of the velocity time curve has been completed, the table would be stopped. Is that clear?

Muller: You would introduce another shock though, wouldn't you, if you bring it to a stop again?

Gareau: No. It would be stopped when the velocity was zero.

* * *

VIBRATION TESTING MACHINES IN THE ONE TO TWENTY TON LOAD RANGE FOR SIMULATION OF TRANSPORT CONDITIONS ON CRYOGENIC EQUIPMENT AND MISSILES

E. H. Brown and R. M. McClintock, National Bureau of Standards,
Boulder, Colo.

Large airborne cryogenic equipments weighing up to 10 tons are subjected to sinusoidal vibration testing on the NBS-60,000 mechanical shaker. The experience gained has been used to design a second machine capable of testing units weighing up to 20 tons.

The vibration laboratory of the National Bureau of Standards' Cryogenic Engineering Division at Boulder, Colorado, was originally constructed to simulate field transport conditions on large, refrigerated, liquid hydrogen and liquid nitrogen containers for the U. S. Air Force. Because of the proximity of gas liquefaction facilities at the Boulder Laboratories, a large shaker was designed and erected at Boulder. This shaker is capable of imparting sinusoidal vibrations to loaded liquefied gas containers called "dewars" weighing up to 12,000 lb over a frequency range of 5 to 75 cps at various required amplitudes. Although operating essentially at constant amplitude rather than constant force, this machine will be called the NBS-60,000 shaker with reference to the peak force developed.

The work performed at the vibration laboratory naturally includes testing of objects of such size as this NBS-60,000 shaker, alone, can handle. Vibration and acceleration tests on lighter objects can also be made with smaller equipment, such as the M-B 2500-lb force shaker. Frequently there are problems associated with vibration testing that are best dealt with by the facilities at the Boulder Laboratories. Such associated problems include cryogenic testing programs, requirements for large quantities of liquefied gases, hazards which arise with the presence of liquid hydrogen and liquid oxygen, and the need for other equipment at the Boulder Laboratories—such as the primary U. S. Frequency Standard—which may be required for special tests. A special telephone line to the nearby Radio Standards Laboratory connects the vibration laboratory with the primary U. S. Frequency Standard which can supply 100 mc or submultiples with an accuracy of one part in 10^{10} for tests on electronic equipment.

DESCRIPTION OF THE NBS-60,000 SHAKER

The NBS-60,000 shaker is located in a concrete-lined pit in the center of the shaker room. The walls of this pit serve both as foundation for the shaker and as blast protection. Test objects are brought into the shaker through the end of the pit by way of a sunken drive.

Test objects weighing up to 12,000 lb, 6 ft in diameter and 18 ft long can be accommodated in the pit itself. Since the pit is open at one end, objects up to approximately 24 ft long can be tested if they can be mounted within a 10-ft section near their center.

The shaker itself consists of a reinforced steel table mounted on coil springs and driven by counter-rotating, eccentric-shaft shaker mechanisms, located on top of the table for vertical motions, and on one end for horizontal motions. The position of the mechanisms is adjustable so that the resultant force can be made to act through the center of mass of the combined system. Hoists are available for lifting test objects up to a bomb rack or other suitable mounts on the underside of the table.

The natural frequency of the unloaded shaker system is approximately 3.7 cps. Thus, above about 5 cps the amplitude of vibration for rigidly mounted test objects remains essentially constant with increasing frequency. In order to obtain variation in amplitude, three different pairs of mechanisms, having total unbalances of 100, 360 and 480 lb-in are available.

These mechanisms can be used either singly or in pairs, and provide six obtainable amplitudes for a given test object. The amplitudes can be reduced by loading the table with dummy weights. Because of the limitations on bearing loads, the higher force output mechanisms are correspondingly limited in frequency range, the respective upper limits being 75, 40 and 10 cps. Because of the method of force generation and the fact that the measured system damping is only 0.003 of critical, the wave form should be very nearly sinusoidal. Experimentally obtained wave forms bear this out, showing no evidence of higher harmonics or distortion.

TESTS MADE EMPLOYING NBS-60,000 SHAKER

In order to reduce accelerations transmitted from bomb-rack to dewar during shock or high-frequency vibrations encountered in transport, a shock-mount system between the dewar support yoke and the suspension hook and sway braces was designed. Transmissibilities of this system were then measured on the NBS-60,000 shaker by using a dummy having the same weight and suspension system as the tactical dewars designed by the Cryogenic Engineering Division for the U. S. Air Force. These tests showed that the shock-mount system, which was designed for a natural frequency of 15 cps with a 6,800-lb loaded dewar, would give sufficient isolation to prevent the transmission of high accelerations.

The hydrogen dewars designed by the Cryogenic Engineering Division are required to keep liquid hydrogen without loss. To accomplish this, a refrigerator—essentially a Joule-Thomson liquefaction system—was designed as an adjunct to the dewar.

This 3,200-lb refrigerator unit was tested on the NBS-60,000 shaker from 10 to 60 cps at amplitudes considered to simulate field transport conditions. These tests indicated satisfactory performance of the shock mount system but served to point out several mechanical design faults. These faults included a bad resonant point of a heavy explosion-proof power breaker and mounting bracket, and a number of hydrogen leaks which developed under vibration in piping joints.

A LARGE SHAKER WITH CONTINUOUSLY VARIABLE FREQUENCY AND AMPLITUDE

One of the difficulties on the Boulder shaker is that variation of amplitude requires shutting down and changing mechanisms. This difficulty has been overcome by a new design in which amplitude can be continuously varied during operation. This new design is very flexible and can be adapted for test loads up to at least 20 tons and a maximum frequency range of about 1 to 75 cps.

A particular application of this design has been adapted for testing 15-ton objects at one-inch, double-amplitude from 1 to 7 cps, and with a constant force of 60,000 lb from 7 to 50 cps. Such a vibration facility would be able to handle objects up to 50 ft long and 12 ft wide.

The eccentric shafts and the mechanisms for this machine are so arranged that, by proper phasing, the amplitude may be continuously varied from zero to maximum. At the same time, above 7 cps, bearing loads may be held constant with constant developed force over the remainder of the frequency range.

DISCUSSION

Christensen, Firestone: How is the suspension of the platform arranged and is it the same suspension for horizontal drive as for vertical or are they different?

McClintock: I assume you refer to the second design I spoke of. The first is accomplished merely by using coil springs, one at each corner of the table. The second is a little unique in that it's suspension is accomplished by means of air springs. I mentioned the frequencies available as one to twenty-five cps. This necessitates that the natural frequency shall be less than one cycle per second.

Each of the two tables shown is mounted on air springs, one at each corner, and there is an interconnecting damping orifice between two sections in each of the air springs. This allows not only the attainment of resonant frequencies of the desired magnitude of less than one cycle but also gives possibilities for variation of system damping. The manufacturers of such air springs assure us that damping will be available from .01 to 0.2 of critical. This allows us to make any necessary adjustments in the stiffness of the vibration machine when it is to be used in a horizontal direction.

The first design mentioned, the one existing at Boulder, has no allowance for variation of stiffness in the horizontal direction. However, since the damping is not variable and there is so very little anyway, the wave forms obtainable from both vertical and horizontal motion are very nearly sinusoidal, and within the frequency range of the recording equipment it is impossible to show any variation.

Dillon, Northrop, Hawthorne: I gathered from your statement that the attachment points of your specimen are coil springs. Would you clarify that for me, please?

McClintock: I'm sorry if I did not make that clear. No. The vibration table itself is mounted, in the case of the first design, on coil springs, one at each of its corners. In the case of the second design, the table mount is on air springs, but the attachment of test objects to both tables will be either by rigid mounts or by shock mount systems as would be used in normal suspension of the test object during field transport conditions.

Dillon: To pursue another point here, have you in your experience encountered a problem of definition of your input in the sense that if you attach it at more than one point at a time you would have such things as unbalance, out of phase conditions, or structural amplification which would tend to confuse the response?

McClintock: My experience is definitely limited, but this has been given some thought, especially in the second design mentioned, inasmuch as a rigid table is not used for mounting a test object. The two tables are independent of one another and an attempt is merely made to keep the driving forces in phase on either table.

You are definitely right in pointing out the fact that there will be out-of-phase motion of the two tables if they are not equally loaded. There are two ways that this might be remedied if it is an undesirable situation. It may not be undesirable because perhaps under the desired shipping conditions for such a container, or missile in a container, the motions encountered in shipment might be such as one encountered under test conditions. But if it were an undesirable situation, it might be possible to equally load the tables by using dummy weights attached to one of them. Then again it might be possible to suspend the system from two points such that equal loading of the tables would occur.

Now, wasn't there a question of definition of input to testing?

Dillon: Yes. Say you have a specification to which you are testing and there are tolerances which have to be met. Would all four corners have to be within the tolerance or would it be legitimate to average the four to satisfy the condition of the specification? Or would this not really be necessary?

McClintock: Yes, it is necessary to consider it because specifications do not necessarily provide for rotational components. The consideration given in the case of the first design which was then in existence, was the capability for relocating the driving mechanisms on the top of the table simply by brute force. First of all, a preliminary calculation was made as to the probable center of mass of the combined test

table and test object. This was a system of one-degree-of-freedom if there was no shock mounting or two if there was one. Sometimes it was necessary to approximate, to try it and if the first approximation failed, you wrestled with those mechanisms a bit and made the necessary corrections so that rotational components were minimized. This was also the case for longitudinal vibrations, but beyond this attempt to minimize the rotational components no attempt was made to evaluate the validity of the test for those other components known to be present.

* * *

LABORATORY SIMULATION OF UNDERWATER EXPLOSIONS

R. F. Mead, Naval Ordnance Laboratory, White Oak, Md.

The design of submarine weapons and equipment to resist underwater explosions is a major problem to naval designers. A machine has been developed which will subject such equipment to controlled laboratory testing. Shock waves and bubble pulses may be properly sequenced or introduced separately. Wide ranges of peak pressure, depth, bubble period, impulse, and pulse shape are obtainable.

Damage to underwater weapons and submarines as a result of explosions is a major problem to naval designers. The increased complexity of such weapons implies a susceptibility to damage from shock loadings. To design resistance against such loadings into the weapon requires trial and error information. In the past, field tests involving full-scale weapons have been the only source of this information. Field tests are, of necessity, not only slow and expensive but yield a minimum of data.

Under joint sponsorship of BuOrd and BuShips, NOL began working on a laboratory facility to provide a means for subjecting weapon and submarine components to simulated underwater explosions. In the process of development several techniques were investigated, some discarded as impractical or incomplete, but one technique seemed promising. This technique led through a prototype machine to the present facility which is called the UWX Machine (Underwater Explosion).

Before further introducing the UWX Machine, it is necessary first to discuss an underwater explosion. When a charge is exploded underwater a pressure wave (compression) is produced which propagates at the speed of sound in water, approximately 5,000 ft/sec. This initial pressure pulse is called the shock wave and is associated with the first expansion of the gaseous products of the explosion. These gaseous products form a globe which will oscillate and upon each expansion produce secondary pressure waves known as bubble pulses. The shock wave is a relatively short-duration pulse of high peak pressure which can be approximated mathematically by $P = P_{max}\ e^{-t/\theta}$ where θ is the time constant and is related to the charge weight.

The first bubble pulse does not attain pressures of the order of the shock wave but does have a greater duration and therefore may exhibit greater impulse. Bubble pulses after the first can usually be neglected. In general, when discussing an underwater explosion, one speaks of a shock wave followed at a later time by a bubble pulse. These pressure waves follow the rules governing elastic waves. Therefore, at a free surface a reflection of the opposite sense occurs and at a perfectly rigid boundary a reflection of the same sense occurs. These reflections play a part under certain conditions of position relative to free or fixed surfaces. One can expect a cutoff or reinforcement of the incident pulse as a result of these reflections (Figure 1).

The UWX Machine was designed to generate a shock wave and bubble pulse which would contain the essential variables present in an underwater explosion. These were considered to be peak pressure, impulse, time constant, and sequencing of pulses. This machine operates first to produce the shock wave which it accomplishes by imparting velocity to a hammer. The

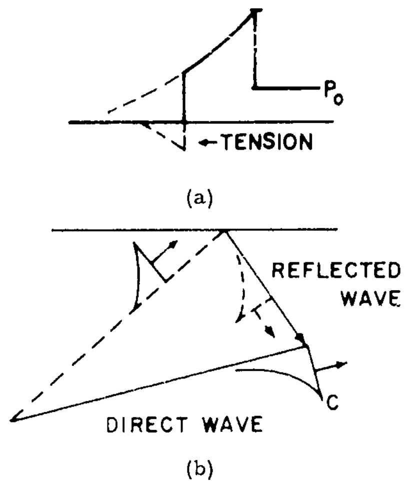

Figure 1 - (a) direct wave and surface reflection, (b) resultant pressure-time curve at point C

hammer is air driven and moves upward until it impacts a piston at the bottom of a vertical water column (Figure 2). This water column may be under prior hydrostatic pressure to a maximum of 2,250 ft (1,000 psi).

The impact transfers hammer energy to the water and results in a rapid pressure rise followed by an exponential decay. The pulse formed migrates upward at the velocity of sound in the water. The time constant governing decay of the pulse is related to the piston mass/area ratio and can be varied by substitution of other pistons. The other parameters are more easily controlled. Peak pressure is a function of hammer velocity and is controllable through the air pressure used to accelerate the hammer.

Impulse variation results from a choice of position of the test item within the vertical water column. In the UWX Machine there is a reflection which occurs when the shock wave reaches the upper end of the water column. Expansion into a reservoir section produces a free-surface type reflection. This reflection being opposite in sense drops the pressure at points it passes to values below the initial pressure. This cutoff provides the means for varying impulse.

Maximum duration and impulse for a given set of conditions, occurs at the piston or bottom end of the water column. The impulse will decrease as a function of position to a zero value

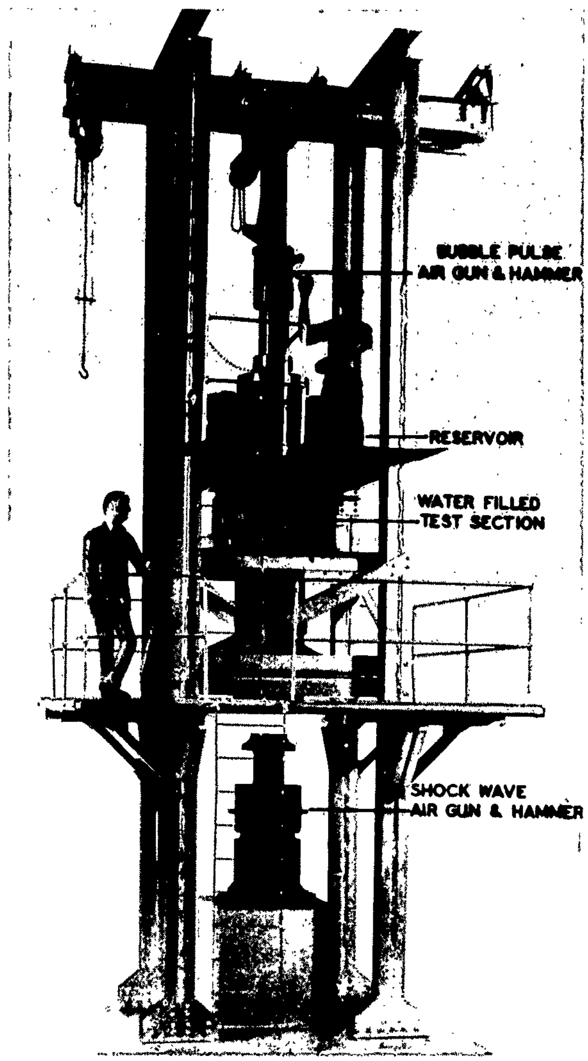

Figure 2 - UWX Machine (1/10 scale model)

at the reflection point. The traveling wave, moving up and down in the UWX Machine, undergoes other reflections as it reaches the ends of the water column and at certain times returns to positive pressures. These secondary positive pulses are smaller than the first pulse but may play a part in the excitation or damage to certain components or systems. These reflections must be considered in interpreting results obtained in the UWX Machine.

Sequencing the shock wave and bubble pulse is quite easy since it only requires a time release of a second hammer. This second hammer driven by compressed air is fired vertically downward and strikes a piston resting

on an air cushion atop the reservoir. The piston produces an adiabatic compression of the air cushion and results in a pulse which approximates the bubble pulse. The bubble-pulse hammer is held by an electromagnet and can be released by breaking the magnetic circuit. A photocell operated by the shock wave hammer starts a timer which can be set to release the bubble-pulse hammer after a predetermined time delay.

The shock wave build-up time was previously stated to be about 300 microsec. This somewhat slow rise is primarily due to a domed hammer. The dome was incorporated to insure central impact regardless of possible malalignment between hammer and piston. A flat-faced hammer is now being prepared and should reduce build-up to a much shorter time.

Specimen mounting is a problem in a facility of this type. When a component is removed from its normal environment, certain restraints are eliminated. These restraints can never be exactly duplicated so in general two mountings are employed such that the test conditions will tend to provide an oversimulation. It is considered that two tests, one with the specimen rigidly mounted and one freely suspended will provide data which will bracket actual field effects. This approach would pertain to items which would respond or be damaged primarily by inertial loading. A device susceptible to pressure loading might not require such consideration.

In the case of inertial loading, acceleration information provides a useful guide for comparing the severity of field and laboratory results. All specimen handling and mounting is accomplished through an opening in the top of the UWX reservoir. The specimen is lowered on a cable (free) or pipe (rigid) suspension which is attached to a framework in the reservoir. The distance of the specimen down into the test section (vertical water column) is determined from the desired impulse or pulse duration.

Tourmaline gages are used to measure the dynamic pressures (Figure 3). There are three locations in the test section wall which provide mounting for these gages. One is not limited to these locations since gages on cables can be placed within the test section but it is not generally necessary to use other locations due to ease of interpolation for determining the pressure history at any point. The gages require a cathode follower type preamplifier to provide an impedance match between the gage and oscilloscope. Much of the recording is done with a single channel oscilloscope and Polaroid camera.

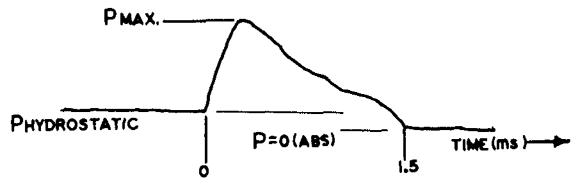

Figure 3 - Typical oscillogram

For operations requiring more channels, the NOL six-trace oscilloscope and 35 mm camera are used.

The prime capabilities of the UWX Machine are:

1. Simulation of initial depths to 2,250 ft (1,000 psi),
2. Shock wave peak pressures to 4,000 psi (30 ft from the explosion of 300-lb TNT),
3. Maximum shock wave duration of 3 milliseconds,
4. Maximum test specimen diameter of 15 inches,
5. Separation or sequencing of shock wave and bubble pulse.

The UWX Machine has only recently been completed and calibration and shakedown tests are still underway. The results obtained to date agree quite well with theoretical performance curves. Certain minor deviations require further investigation to establish the associated causes. Minor design changes are in progress to improve the operation of the machine and to improve the simulation it produces. Some tests of ordnance components have been conducted with encouraging results which show good correlation with field data.

The types of investigations which can be carried out are those combining the variations of initial selected simulated depths with selected shock-wave peak pressures in one of the three distinct types of studies as follows: (1) shock wave effects alone, (2) bubble pulse effects alone, and (3) the combined effects of shock wave and bubble pulse.

Probably the most significant conclusion is that there now exists a facility which can produce a reproducible pulse bearing a marked similarity to the shock wave developed by an underwater explosion. This facility also permits ease and speed of data collecting with reference to damage or response of ordnance and submarine components resulting from underwater explosions.

DISCUSSION

McKeehan, ESL, Yale Univ.: My interest at present is with the Electric Boat Division of General Dynamics Corp. The problem of simulating shock on supported objects inside a hull, passed over in the last presentation, appears to be open to attack by the same method. The author is dealing with an open pipe and considering the pressure variations within the open pipe before the reflection from the free end gets back or before the combination of the positive and negative pulses. All one would have to do would be to terminate the pipe with a plate representing the mounting surface in the ship's hull. This might mean that you would have to enlarge the section as it went up and accept some attenuation of the total impact per unit area. But it would still be possible to shape the pulse received as you do here, and also the reaction of the mounted element, so as to simulate in a pretty good fashion the shocks on hull mounted structures. I merely wanted to point out that the possibilities of this type of experimentation have only just been started.

Mead: That was a very good point you brought up. The machine seems to have a large number of possibilities. Dr. Bryant, whom I mentioned before, is planning to deal with scale models of submarines. I have looked into the problem of scaling a little and it bothers me somewhat. I have to get in touch with Dr. Bryant to see what he intends to do. It would seem that in order to scale, we have to adjust the speed of sound beyond easily scaled limits. The transit time across any scaled-down model is going to be correspondingly faster because of the small size. I believe the transit time should scale as the square or the square root—I am not sure which at the moment. I have not found a good way yet though I do think it is possible. If you just wish to scale the impulse, the machine will scale very nicely.

Taylor, Portsmouth Naval Shipyard: It was mentioned that you could make a machine with a steel plate on the end instead of an open chamber. Such a machine was made at the request of the Bureau of Ships many years ago by Portsmouth Navy Yard because that is a problem that is very well recognized in submarine design. Every component in the hull must be at least as strong as the hull and be able to take pressure transients of this sort.

The machine at the Naval Ordnance Laboratory provides a shock similar to that of the Portsmouth machine. We do just what was suggested. We have a closed chamber with a piston which is struck by a moving weight very similar to this machine. We get pulses similar in duration and of somewhat larger intensity than the shocks developed by this machine. The Portsmouth machine has been described at these meetings; the paper was published in Shock and Vibration Bulletin No. 17.

Dranetz, Gulton Mfg. Corp.: For scaling down, I might suggest the use of other media surrounding the actual model which might have a completely different velocity of sound.

Mead: I think alcohol drops the speed of sound down to about 2,500 ft per sec. I am not too sure at the moment, as I said, whether we want to scale time up or down. If we wanted to go up we could probably get mercury. It's a little impractical, but nevertheless possible. It just does not seem that the range is still there but would fall short.

Mains, KAPL, General Electric Co.: I have a question for Mr. Mead on the matter of the smoothness of the shock pulses that he showed. What was the frequency response of your pulse measuring devices as opposed to the frequency of the pulse? If there had been hash in it, would you have seen it?

Mead: I am by no means an electronics man. I know that the gauge and instrumentation is sensitive to at least 10,000 cycles. It would seem from a theoretical standpoint that there would be some hash. You have an elastic wave running through the steel piston which should produce something to show up on the record. What the frequency of these would be I don't know. I haven't actually calculated this and it has bothered me a bit. I don't know just why it hasn't shown up on the record. One might also expect other hash, possibly reflections, off the sides of the system or off our gauge mounts. I just feel that these reflections would show up on the record if they existed because I believe the instrumentation is of high enough frequency to record them. But we have not seen any sign of them.

* * *

A NEW 300-FOOT UNIVERSAL DROP TOWER

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This paper describes the new 300-ft universal drop-test tower which is now being installed at Sandia Corporation. The tower will provide a means for drop-testing materials and instruments at high-impact levels under controlled conditions. The problems involved in developing the various mechanical and electrical components are discussed.

A new 300-foot drop facility is currently under construction by Sandia Corporation at their outdoor test area in Albuquerque, New Mexico. This paper gives a general discussion of the tower installation, including a brief discussion of some of the design considerations involved.

BASIC CRITERIA

Work is currently going forward on an instrument recovery program. This program has as its goal the eventual development of a series of drop-test instruments which will be operable not only during the free-fall phase of a drop, but also during the actual impact phase at the end of the drop. Similar studies are being conducted by most of the major aircraft companies. To determine the feasibility of Sandia Corporation's specific program, a series of high-velocity ground drops were conducted using the temporary installation shown in Figure 1. This photograph shows the 100-ft crane, down-line, release mechanism, and target. The pole shown in the foreground is used to guide the necessary instrumentation lines.

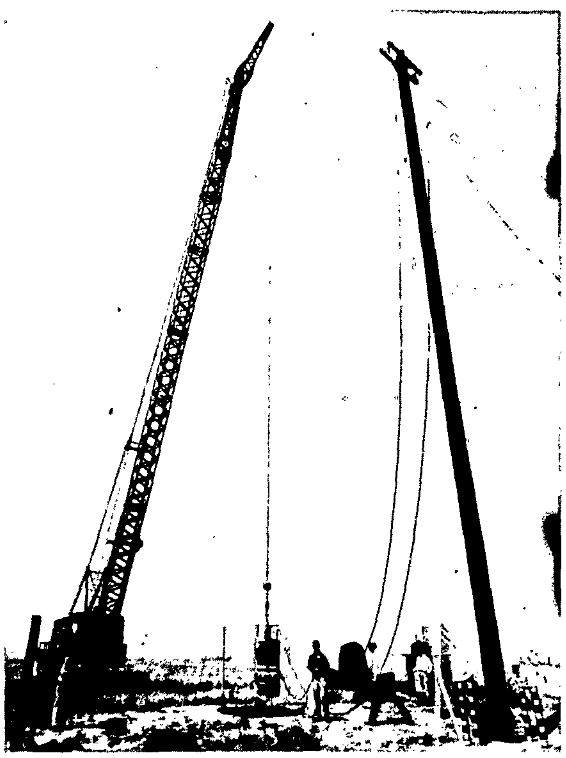


Figure 1 - Temporary drop-test installation

The new 300-ft tower was requested so that the range of these tests conducted with the crane could be extended. A capability of up to 130 ft/sec vector velocity with horizontal velocity components of up to 60 ft/sec was requested as the basic requirement for the new tower. The 130 ft/sec requirement dictated a tower

height of about 300 ft since 300 ft corresponds to a terminal velocity of 139 ft/sec. The request for horizontal velocity components dictated use of a track or guide to bring the test unit down along an inclined path. To avoid extraneous instrument indications, it was necessary that the test unit be falling free at the time of impact. Test-unit weights of up to 3,500 lb were specified.

The results of these various criteria were combined into a design as shown in Figure 2. In this view the moving drops are conducted by sliding a carriage with the test unit attached down the 45° cable shown at the left of the picture. At the proper instant, the carriage releases the test unit, and the test unit continues downward on a free trajectory into the target slab. To provide structural symmetry, the same cable array is duplicated on the right-hand side of the tower. This side, however, is not equipped for running drops but only provides for directly vertical drops into a single square target slab. By utilizing both sides of the single tower in this manner, near-simultaneous drops can be made on the two sides with the moving or dynamic drops on the left, and the static or vertical drops on the right. A single set of operational equipment and instrumentation will suffice for the two setups. It is this provision for both dynamic and static drops which has caused the installation to be termed "universal."

TOWER—GENERAL

The tower is 300 ft high, 3 ft square in cross section, and is constructed primarily of 4 x 4 x 5/16 steel angles with suitable 2-1/2 x 2-1/2 x 1/4 angle bracing. It weighs approximately 10 tons. Lateral stability is provided by sixteen 3/4-in guy cables spaced as shown in the figure. Each cable carries a 6,000-lb tensile load with a safety factor of 2.5. The tower is capable of withstanding a wind greater than 100 mph.

Access to the top of the tower is by means of a caged ladder which spirals around the tower with safety landings every 30 ft. The tower is surmounted by a work platform, special track cable support, two hoisting cable sheaves, and aircraft warning lights.

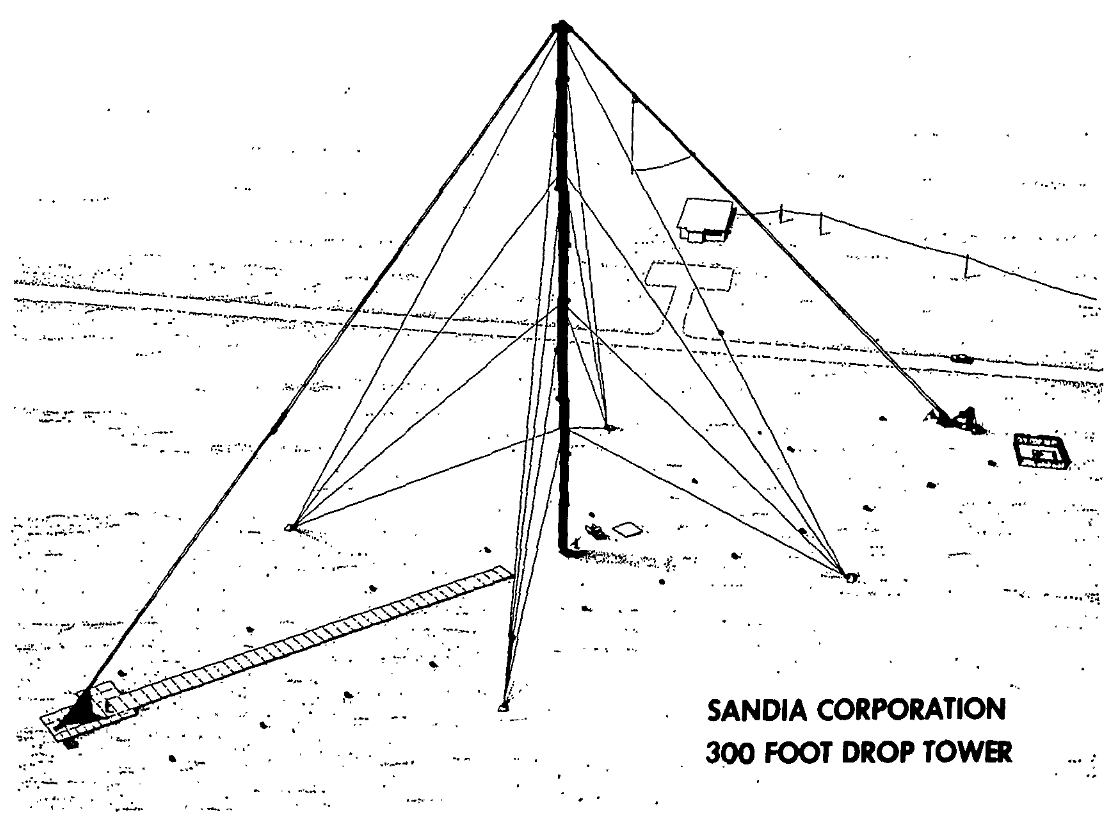


Figure 2 - The new 300-ft drop-tower installation

The total area covered by the tower and its associated cable array is over 10,000 sq yd.

TRACK CABLE

The main or track cable which supports and guides the carriage and drop-test unit down the required angular path is a single continuous piece of smooth surface, 1-1/2-in steel Lock-coil cable 820 ft long. A smooth surfaced cable is used to reduce initial instrument vibration and to reduce carriage friction. The cable is preloaded to 20 tons and has a breaking strength of 93 tons. The relatively high preload is necessary to provide semirigid cable behavior during operation.

The saddle at the top of the tower which supports the track cable is roller-lined to permit some relative motion between the track cable and the tower and to give the necessary cable bending radius without the bulk and weight of a 12- to 14-ft pulley.

HOISTING CABLE

To hoist the test units up to the proper drop height, special permanent hoisting equipment had to be provided. A continuous-loop system was devised, consisting of 1,600 ft of 9/16-in steel cable located 15 in above the main track cable on both sides of the tower. The cable return between the two track ends is underground. The underground portion is supported on suitable support sheaves and is protected by a ceramic conduit. Us^ of the continuous-loop system precludes asymmetrical hoisting cable loads on the tower and eliminates the need for more than one set of hoisting drive units.

TARGETS

Two separate targets are provided for the two drop sides. Directly under the track cable on the static drop side is a single concrete pad 12 ft square and 2 ft thick. This is the target for vertical drop testing. On the dynamic drop side is a full-length concrete target 260 ft long, 12 ft wide, and 2 ft thick. Both targets are thoroughly reinforced with steel rods near both the upper and lower surfaces of the concrete. Four "portable" steel plates each about 6 ft long, 12 ft wide, and 6 in thick are provided to protect the surface of the concrete at the predicted points of impact. This protection is necessary since the maximum impacts expected involve over one million ft-lb oı energy. It is hoped that the plates will distribute the load sufficiently to prevent surface cracking and spalling of the concrete. The targets have been poured in sections so that it is possible to replace them if the surface protection proves to be inadequate.

The small target weighs 22 tons; the larger one, 468 tons. The steel facing plates weigh an additional 35 tons, thus making a total target weight of 525 tons. The total weight of concrete required for the project is 800 tons.

STATIC-DROP SIDE

The equipment at the static-drop side anchor point is shown in Figure 3. In this photograph, the track cable is shown coming down from the upper right and terminating at the block and tackle. The block and tackle and its hand winch are in turn attached to the main counterweight arm.* This 4-ton counterweight and lever arm maintain the necessary 40,000-lb track cable tension, compensate for changes in cable length due to cable "snap" at the instant of drop, and correct for thermal expansion of the tower and cable system as the ambient temperature changes through the usual 24-hour cycle. The block and tackle and hand-winch combination compensates for progressive cable stretch or creep and for wide-range thermal-expansion changes such as are experienced between winter and summer. The main counterweight is limited in travel to a practical amount by the use of a stop-frame with rubber snubbers. Undesirable counterweight oscillation caused by either drop action or wind resonance is eliminated by an aircraft-type shock absorber attached to the counterweight arm.

The hoisting cable in this photograph also comes down from the upper right and is seen just above the track cable. At its lower end it is turned to one side to avoid the main counterweight. It then makes two complete

*The counterweight shown in this photograph has been broken and moved over to permit visual inspection of the motor installation at the rear. The counterweight on the actual installation is intact, of course.

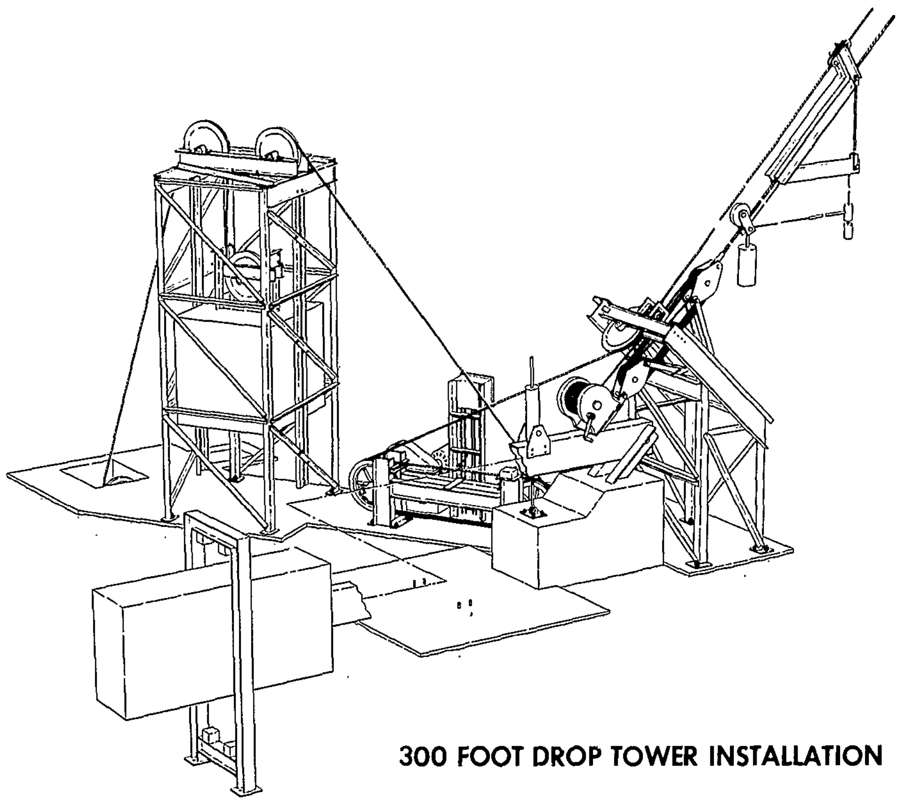


Figure 3 - Detail of the static-drop side

passes around the hoisting sheaves to obtain the necessary hoisting traction, passes through the hoisting cable counterweight tower, and then goes underground. As previously stated, it travels underground to the far end of the dynamic-drop side, up the dynamic-drop side, over the top of the tower, and back down the static-drop side, thus making a complete circuit. The hauling cable counterweight maintains a 5,700-lb tension, thus effectively preventing tangling between the two adjacent operational cables.

The hoisting gear consists of two duplicate 3-hp units, each consisting of a motor, speed reducer, drive sheave, and solenoid brake. A revolution-counting switch protects the hoisting array from excessive overtravel in either direction. Hoisting-motor control is accomplished by means of a plug-in control unit with plug-in points at the base of the tower and at the track cable ends. A hoisting speed of 50 ft/min provides a compromise between speed and safety and permits a round trip up the tower and back in about 12 min.

The static drop carriage, shown in place on the track cable, is a 5-ft long aluminum channel assembly arranged to slide on the track cable. It is attached securely to the hoisting cable. Mounted horizontally on the underside of the carriage is a standard twin-lug aircraft bomb rack. A down-line and antitwisting cable are provided for use in drops at less than maximum height. An explosive device on the lower end of the down-line provides a release mechanism for these short drops.

MOVING DROP SIDE

The moving or dynamic drop side anchor assembly (Figure 4) is a much simpler installation, and for good reason. On this side, test units will ricochet off the target, in line with the anchor point, and hence no hoisting or similar equipment can be installed here. The only two pieces of equipment at the anchor point are the carriage-arresting gear or bumper and the barricade. In normal carriage operation, some moderate impacts at the lower end will occur. In the case of accident or long free-runs, serious impacts will occur.

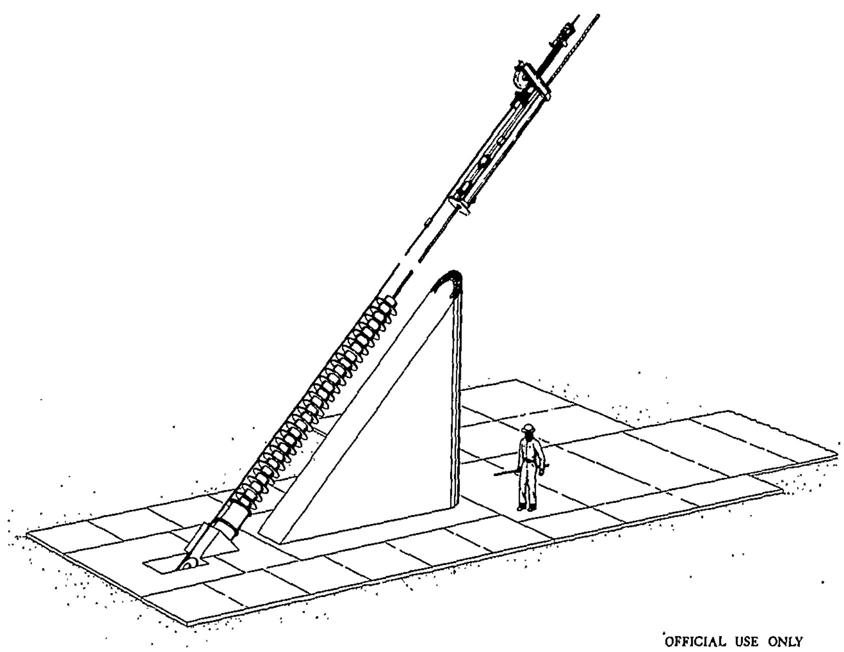


Figure 4 - Detail of the moving-drop side

To minimize the effects of either type of impact on the carriage, a giant rubber bumper has been devised. It has an effective length of 15 ft. The bumper consists of 10-1/2 ft of quite hard, 60 Durometer rubber. The rubber is topped by 2 ft of high-hysteresis foam plastic, 2 ft of somewhat softer 40 Durometer rubber, and a final cap of 6 in of industrial sponge rubber. The sponge and the 40 Durometer rubber absorb the initial-impact load. The 60 Durometer rubber carries the major portion of the load on the bumper. The foam plastic adds the necessary damping to the system.

All of the rubber materials are used in the form of disks 10 in in diameter and 1 in thick. The disks are slit radially to permit attachment to the track cable. Slits in adjacent disks are staggered 120° to preclude the erratic operation which would result if they were aligned. To prevent failure due to progressive maldistribution of the load, light but stiff oversize aluminum disks are interspersed in the bumper with approximately one to every 8 to 10 in of bumper length.

The barricade appearing in this figure is also an emergency item. It is provided to protect the track cable anchor point from possible test unit impact. It is a rugged concrete and steel pylon mounted on an oversize foundation. The face is surfaced with heavy railroad rails laid side by side and welded together. The rails serve to deflect the stray test units, and the mass of the combination, over 108 tons, serves to absorb any direct impacts.

MOVING DROP CARRIAGE

Design of the carriage or cable car for the moving-drop side was an especially interesting problem. It was required that the cable car be capable of sliding down the track cable with a minimum of friction, that it be long enough not to pitch as a result of cable oscillations, that it be strong enough to easily sling a 3,500-lb test unit on its underside, that it be rugged enough to withstand impacts of 100 to 200 g, and that it be light enough so that an economically feasible arresting gear could hold it at the 100- to 200-g limit. It was desirable that the carriage itself have automatically engaging cable brakes to aid in the arresting action. In addition, the carriage release unit for carrying and dropping the test unit had to be fast-acting, positive, rugged and/or replaceable, and if possible a secondary or back-up drop unit release action was to be provided.

To keep the terminal impact down, a 500-lb maximum-weight limit was set on the proposed carriage. As a result of preliminary model tests, a carriage length of 10 ft was selected as

being as short as permissible if adverse pitching were to be avoided. To assure strength and longitudinal stiffness, the carriage was constructed from two 8-in aluminum channels spaced 3 in back to back with the main-track cable passing between the two.

Wheels to permit motion along the cable were first considered as the most logical means for minimizing friction, but they were abandoned when rough calculations showed that wheels of the required minimum size and weight would have to achieve 5,000 rpm in less than 4 sec. Slides or slippers were therefore used. These are made of bronze and are constructed so as to be readily replaceable. A hexagonal shoe groove is used to provide clearance.

Design of the brakes was also a problem. The entire braking unit could not weigh more than 150 lb, and there could be no trailing cables or hoses. A hydropneumatic accumulator was finally selected to power the brakes, and by using pressures of from 2,500 to 3,000 psi it was possible to keep the size and weight of the braking system within reasonable limits. The final design employs a 1-pint accumulator. A single brake shoe is used in conjunction with the regular rear shoe to provide the necessary braking action. A small hydraulic ram powers the brake shoe.

The brakes are energized by either the release of the drop unit, that is, on pull-out, or, as a secondary measure, whenever a clip-like device intentionally placed on the secondary cable is struck by a mating fork provided on the top of the carriage. In this way, the brakes normally engage just after the test unit is released or, if release does not occur properly, the emergency clip will trip the brakes. The clip is a frangible steel disk which is spring-loaded at its outer edge. It is about neutrally stable so that any impact with the brake-actuating fork causes it to disintegrate mechanically and thus avoid impact with the carriage pulley.

A bleed-down orifice is provided in the hydraulic system to cause the brakes to release automatically after a predetermined period, about 2 min in most instances. This allows the carriage to come down the cable after the braking action has been completed.

The moving-drop carriage has two separate units for supporting the drop unit. One is a conventional twin-lug, 30-inch bomb rack which is used for small drop units and large drop units at normal attitude. The other consists of special cable attachments at the front and rear of the carriage. These attachments contain explosive-powered, electrically triggered cable cutters which, when required, simultaneously cut the front- and rear-drop unit attachment cables. Both types of drop-unit release have backup release systems to assure positive drop action.

To hoist the carriage up the track into position for release, the carriage is attached to the hoisting cable rider by means of a light-cable loop. The cable rider also has an explosive cable cutter which at the desired moment cuts the cable loop and allows the carriage to start its trip down the cable.

The test unit release point is determined by an electronic timer. Initial motion of the carriage breaks a circuit which starts the timer. The timer runs for a preset interval and then turns on the power to release the drop unit. By varying the initial location of the carriage and the interval setting on the timer, a wide range of vertical and horizontal velocity combinations are available.

PRELIMINARY STUDIES

Before the tower installation was designed, mathematical studies of the catenary envelop were made. Mathematical studies of the dynamic cable behavior were also attempted; however, the complexity of the mathematics encountered here dictated a series of similitude tests.

For these tests, a 1/26 scale model was used (Figure 5). All sizes, weights, and accelerations

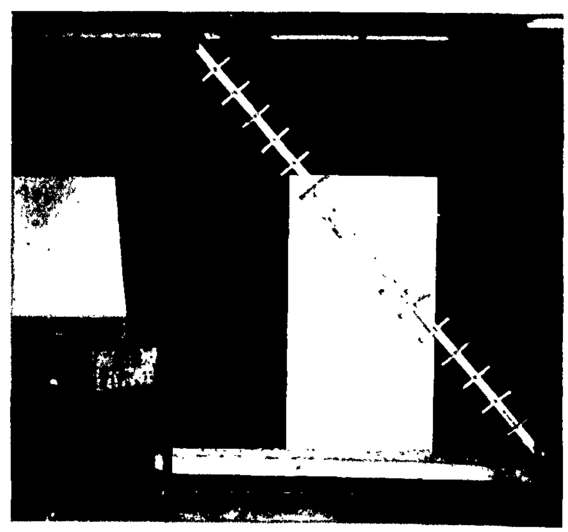


Figure 5 - Similitude model of the moving-drop side cable installation

were properly scaled, and photographic studies of the cable operation were made. As a result of the scaling laws involved, the model cable oscillated 5.1 times as fast as the full-scale cable. The movies were therefore shot at 5.1 times the normal speed. Then by projecting the model movies at normal speed, the dynamic behavior of the full-size cable could be witnessed. From these model tests if was possible to determine the carriage length, to validate the cable tension, and locate the position of the haul cable.

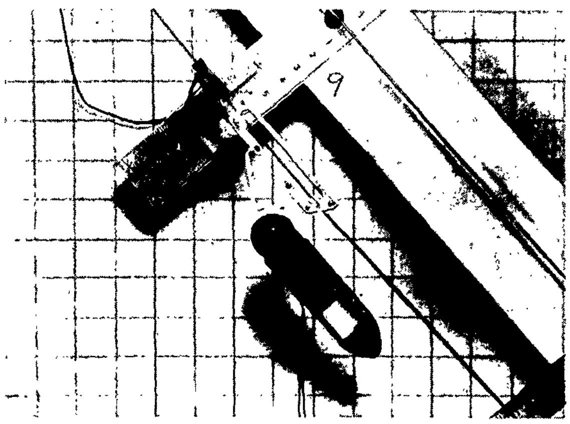


Figure 6 - Detail of the model-drop carriage

Close-up movies were taken to establish cable oscillation amplitude and possible test unit pitch at the instant of release (Figure 6). Maximum cable amplitude was established as 7 ft measured normal to the cable. The movies also confirmed the direction of the force field. Before a drop, the force field is vertical. As soon as dynamic stability is established, after the carriage has been released but before the test unit is dropped, the effective force field has shifted to a direction normal to the cable. Thus the bomb racks on the moving carriage had to be installed parallel to the carriage axis to assure proper separation at the time of release.

CONCLUSION

The tower facility is scheduled for completion by April 15, 1956. Initial operation of the facility will establish whether the various engineering compromises incorporated in the design are correct. Since design and construction of the tower has been on a crash basis in a field where little previous engineering data were available, it is probable that some design changes will be necessary. However, based on calculations and model tests, it is expected that the general performance of the installation will be more than adequate.

DISCUSSION

Stooksberry, Case Institute of Technology: I am wondering what angle of impact it has been designed for? Apparently this is not a variable.

Westgate: Yes, it is a variable. We designed it so that the whole installation will be capable of dropping the units to land in the exact attitude in which they are released. The units are located as shown in the model and they can be swung through 45 degrees to bring them to vertical and a further 15 degrees from that. As a result we have a total of 60 degrees of variation in possible attitude. We hope that by the care exercised in the release mechanism we shall avoid tumbling of the item on release.

Sowel, Hollowman Air Development Center: There was no mention of maximum g expected from this test machine?

Westgate: I think the maximum g is a completely open deal. We hear discussions in terms of several thousands, but are going to wait and see what we get. We have a velocity of 130 ft/sec. Your guess would probably be as good as mine.

Sowell: I'd guess 1,000 or more—2,000 is probably right.

* * *

THE HYGE ACTUATOR IN SHOCK TESTING

J. B. Ottestad, General Dynamics Corp. (Convair)

The paper describes a new type of gas-energized, hydraulically controlled, accelerating device. Stored energy in the form of compressed air is released instantaneously, and the waveform is then controlled by means of hydraulic flow through an orifice controlled by a metering pin.

INTRODUCTION

With the new developments that are being made in aircraft, missiles, combat ships and other ordnance the problem of shock is multiplying. Not only are the numbers, types and masses of the components increasing but also the very nature of the shock is expanding to cover wide ranges of peak-acceleration level and waveform.

In view of this expanding problem it is no longer practical to limit shock tests to final maximum environmental conditions as applied to full assemblies. In addition, testing of components for the full-shock spectrum leading to the maximum conditions must be conducted in order to determine the shock-resistance ability of the component.

To meet these conditions a new type of shock-testing device had to be developed. A device that would not only be capable of meeting the wide range of conditions required, but also be practical from the standpoint of size and cost so that the small manufacturer could conduct component evaluation for shock conditions while that component was still in the developmental stages.

BASIC PRINCIPLE OF HYGE

As a direct result of a program aimed at the solution of the shock simulating problem Convair has recently developed a new high-thrust device for shock testing. Designated the "HYGE Actuator" (pronounced Hi-gee) this device imparts high, accurately controllable thrust to a test specimen by means of the instantaneous release of stored energy in the form of compressed gas.

The two basic problems in this type of unit have been solved. The first is the development of a simple, reliable, and predictable method of instantaneously releasing the compressed gas. The second is the development of a means of controlling the ensuing high rate of expansion of the gas in a manner that allows the developed thrust to be readily and accurately controlled.

RELEASE PRINCIPLE

The HYGE accomplishes rapid activation by a pressure-thrust amplifying principle. The thrust developed by a high pressure acting over a small area must overbalance the thrust of a low pressure acting over a large area. When this occurs the high pressure is allowed to react also over a large area, giving a great unbalance thrust. This principle can be more clearly shown in reference to Figure 1.

The HYGE is basically a cylinder separated into two chambers (noted as "A" and "B") by means of an orifice plate.

Riding in chamber "A" is a piston connected to a column which in turn reacts against the

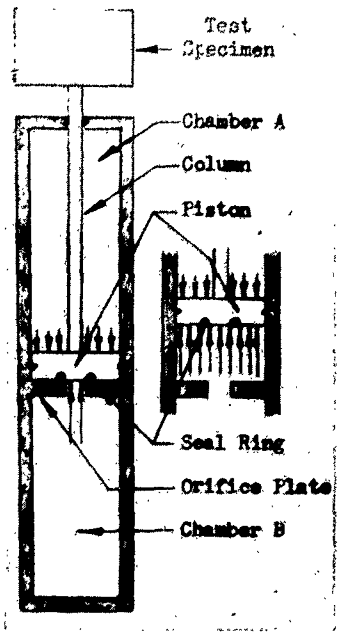


Figure 1

test specimen. On the under face of the piston is mounted a seal ring.

When chamber "A" is pressurized to some low value (P_1) there is developed a resulting downward thrust on the piston equal to P_1 times the top area of the piston (A_1). This thrust forces the piston against the orifice plate allowing the seal ring to make tight contact around the periphery of the orifice. Thus the underface of the piston, with the exception of the small area inside of the seal, is isolated from chamber "B."

Chamber "B" can now be pressurized. To obtain a balanced thrust condition the value of this pressure P_2 is very high in comparison to P_1. P_2 is allowed to react on the piston only over the small area inside of the seal ring.

As P_2 increases above this balance pressure, a slight unbalanced upward thrust is developed. The piston responds to this force and starts to move away from the orifice plate. After the piston moves a very short distance, however, the seal loses contact with the orifice. Immediately, the high pressure P_2 reacts, not only on the small area inside of the seal ring, but rather on the entire underface of the piston developing a great unbalanced thrust against the test specimen as the piston is forced away from the orifice plate.

It can be noted that by the very nature of this simple principle there is assured absolute repeatability in the developed thrust.

ACCELERATION CONTROL

While the foregoing discussion of the HYGE operating principle indicates the triggering technique to achieve instantaneous release of the compressed gas it is necessary to add control so that any given acceleration-time pattern that falls within the available energy potential can be produced. This control is accomplished by means of hydraulic fluid, that must pass through the orifice. By regulating the flow area for the fluid, predetermined pressure drop can be developed, thus controlling the net pressure that reacts against the piston after activation. This principle is shown in reference to Figure 2.

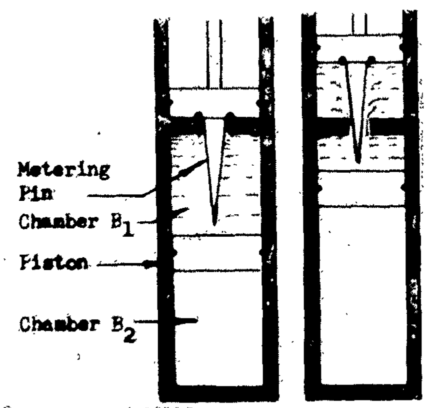


Figure 2

A free-sliding piston is inserted into chamber "B" dividing it into two chambers (noted as B_1 and B_2). To the thrust piston is attached a contoured plug (metering pin) that projects through the orifice. Chamber B_1 is filled with hydraulic fluid while chamber B_2 is free to be pressured with gas.

The overbalancing triggering technique in this situation is the same as previously described. However, upon activation, the hydraulic fluid must flow through net orifice area in order to react against the piston. This net orifice area is a function of the gross orifice area minus the area of the metering pin. Thus the metering pin contour dictates the exact pressure drop experienced by the fluid as the piston is forced away from the orifice plate.

This means of control is highly accurate and reproducible. It also allows wide variation in the acceleration wave form produced by the HYGE unit by simply changing the metering pin.

DECELERATION CONTROL

In many cases a shock specification requires a two-phase pattern. This pattern would consist of a controlled acceleration of high magnitude followed by a controlled deceleration of high magnitude. The acceleration phase has been described; the deceleration phase, however, presents a new problem.

At the conclusion of the acceleration phase the thrust piston, column, and test specimen have reached a maximum velocity. The deceleration phase, therefore, is simply supplying a regulated stopping force. Once again the use of hydraulic fluid being forced through a regulated area provides the solution. Figure 3 shows the general configuration of the deceleration phase control elements.

Chamber "A" is divided into two chambers (noted as A_1 and A_2) by a second orifice plate. A contoured plug (metering pin) is attached to the top of the piston. Chamber A_2 is partially filled with hydraulic fluid as shown in Figure 3.

After the unit has triggered, and at the conclusion of the acceleration phase, the piston has moved upward to a position that starts to force the fluid in chamber A_2 through the deceleration orifice. As the fluid moves through the orifice the net flow area is reduced by the metering pin. This creates a back pressure against the top face of the piston providing the required stopping force.

By changing the metering pin it is thus possible to produce a wide variety of deceleration pulses.

OPERATIONAL ENVELOPE

In establishing the actual physical output of a device embodying the above principles it can be noted that the piston assembly is under the action of basically two opposed forces. These forces are produced by the load gas pressure reacting against the bottom of the piston and the set gas pressure acting against the top of the piston. At the instant of activation the bottom pressure is many times as great as the top pressure. With the reaction areas approximately equal the upward thrust developed greatly exceeds the back thrust. However, as the piston is displaced under the unbalanced force thus produced, the lower gas is expanded causing a pressure drop and the upper gas is compressed causing a pressure rise. The degree of this change is dependent on the physical configuration of the given unit involving the initial gas volumes, pressures, and stroke of the piston.

A plot of the summation of produced forces versus displacement is called the operational envelope. This is actually the maximum potential thrust of a given HYGE at each position in its stroke. An example of an operational envelope for a 3-in diameter HYGE Actuator with a load stroke of 7 in and a 2,000 psi load pressure is shown in Figure 4.

Any acceleration-time pattern as applied to a given specimen mass can be replotted as thrust versus displacement. By comparing this required curve against the operational envelope,

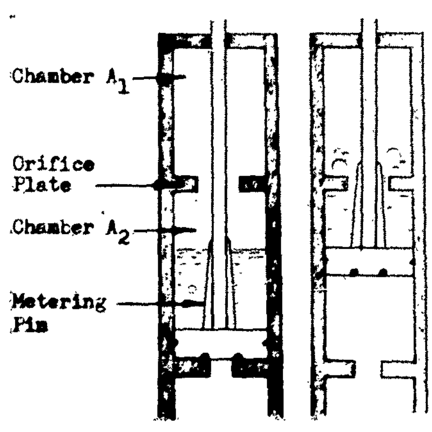


Figure 3

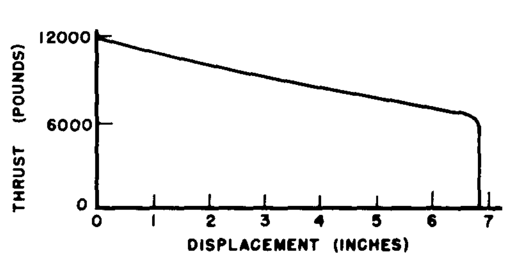


Figure 4

the excess thrust to be controlled by the acceleration orifice and metering pin becomes apparent.

THE HYGE CONFIGURATION

In the development of the HYGE Actuator several units were constructed. One of these, called the 3-in Standard, is shown in Figure 5 with the sectional view shown in Figure 6.

The physical specifications of this unit are:

Internal bore - 3 in.
Maximum piston stroke - 7 in.
Rated operating pressure - 2,000 psi.
Rated output thrust - 10,000 lb.
Overall length - 46 in.
Gross weight - approximately 200 lb.

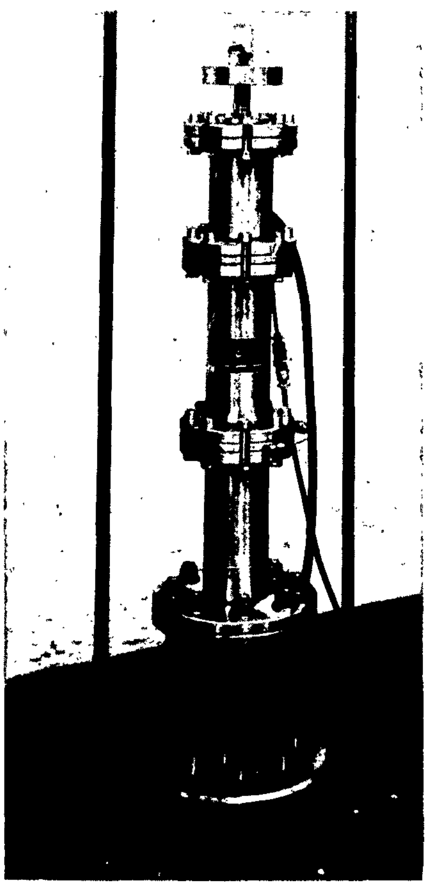


Figure 5

USES OF THE HYGE

During the evaluation of the HYGE units a variety of possibilities were discovered that increased their already wide potential versatility.

High Acceleration-High Deceleration

For shock conditions involving high-level acceleration followed by high-level deceleration, the HYGE was mounted vertically with the test specimen secured to the thrust column.

The actuator itself was attached to a sturdy foundation that could withstand both tension and compression forces. Power for the unit was furnished by a standard 2,200-psi nitrogen tank. Pressure control was maintained by a simple control panel with gauges to check the top and bottom pressures.

High Acceleration-Low Deceleration

For shock conditions that required a high-level acceleration followed by a very low-level deceleration, it was found that the HYGE could be operated either vertically or horizontally as shown in Figures 7 and 8. In this operation the test specimen was mounted in a light carriage that slid along a simple rail system. At the conclusion of the acceleration phase produced by the actuator, the carriage and specimen parted from the thrust column. Deceleration of the carriage was accomplished by gravity in the vertical installation and by friction in the horizontal installation.

Impact Shock

Certain shock specifications require an impact-type loading that will produce very high accelerations (2,000 to 5,000 g) of only momentary duration.

In attempting to simulate these conditions the HYGE was fitted with an impact head or anvil of a given weight (20-200 lb). This was attached directly to the thrust column. The test specimen was suspended a few inches above this head. Triggering the HYGE accelerated the anvil such that upon contact with the specimen the head had a velocity approaching 800 in per sec. Thus impact shocks were produced with a limited degree of control (Figure 9).

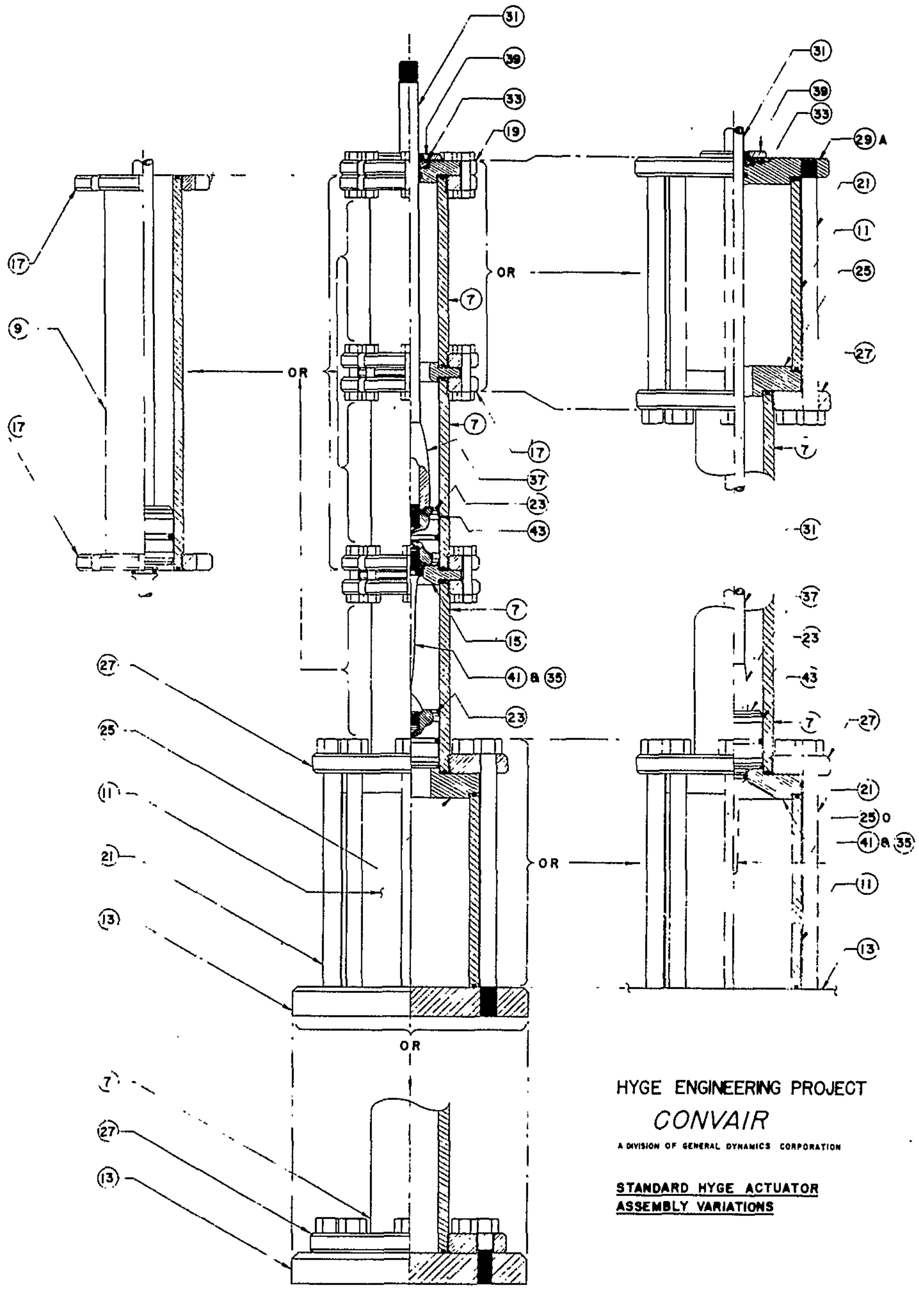


Figure 6

Multiple Units

With a rated thrust of 10,000 lb the single 3-in HYGE was limited to those shock conditions that fell within this capacity. It has been shown, however, that two or more units can be operated simultaneously against a common table.

HYGE TEST RESULTS

In view of the operating principle of the HYGE Actuator, it is obvious that there are an infinite number of possible waveforms, acceleration levels, specimen weights, and shock durations obtainable in one unit.

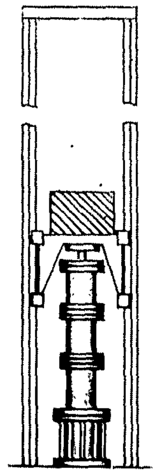


Figure 7

Repeatability - The repeatability or ability to reproduce a waveform is within 1 percent.

Peak Acceleration - The maximum acceleration obtainable on a pure thrust condition has not been determined due to inadequate instrumentation. The maximum recorded has been approximately 300 g.

Versatility - This is difficult to define. However, the following notes were made:

(1) The development of different waveforms can be readily achieved by changing only the metering pin.

(2) With a given metering pin a wide range of curves can be developed with the same basic waveform and differing peak accelerations by simply changing the gas pressures.

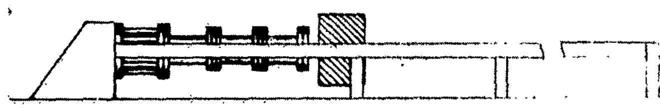


Figure 8

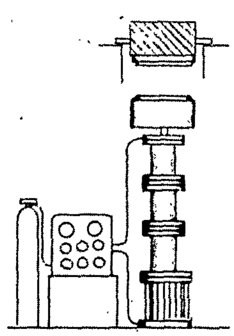


Figure 9

Testing Rate - In instances involving little change to the specimen the rate of testing has been 100 shocks per hour.

The evaluation of the 3-in unit concentrated on three basic waveforms: the 1/2-sine pulse, the square pulse and the 1/4-cosine pulse. The basic peak acceleration levels were investigated (40-100 g) as applied to specimen weights of 40 lb and 80 lb. As a result of this evaluation certain basic characteristics can be noted.

Build-up Rate - The rate of build-up from zero to peak acceleration on the 1/4-cosine pulse and the square pulse reached 200,000 g per second. On a 100-g shock pulse the build-up time is something less than 1 millisecond.

REVIEW

Finally, several points warrant particular note:

Exact Operational Evaluation - The basic principles of the HYGE lend themselves to almost an infinite number of possible conditions, therefore it was not feasible to check them all. Three basic waveforms have been evaluated. These are the 1/2-sine pulse, the square pulse, and 1/4-cosine pulse. This evaluation has been limited to one or two assumed values of acceleration on three or four different specimen masses. However, this test work has demonstrated that certain basic and simple procedures can be followed to develop any new condition required of the HYGE Actuator.

Other Units Sizes - The discussion has centered on the 3-in HYGE Actuator because analysis of existing shock specifications for Convair indicated that this unit, with a 10,000-lb output thrust, would meet most of the requirements. However, there are no foreseeable size limitations of the HYGE Actuator other than economics. For example, a 16-in unit, 200-in long has been designed and is under construction. This HYGE Actuator will have a rated output thrust of 300,000 lb and a maximum stroke of 60 in.

While the HYGE Actuator is not, in its present form, the general and complete device for all shock-testing problems, it does present an inexpensive unit for providing controlled shock over a wide range of conditions.

* * *

PNEUMATIC-HYDRAULIC IMPACTORS FOR PRODUCING CONTROLLED ACCELERATIONS

S. P. Sanders, Bendix Aviation Corp. Missile Facility, Mishawaka, Ill.

This paper describes the design, operation and performance of the pneumatic impactors used to simulate the boost phase of the flight of the Talos missile. The simulation covers the high acceleration at the start of the boost phase and the moderate deceleration between the booster burn-out and the missile's combustor ignition.

During the early stages of the Talos Missile program, it was decided that an impactor would be required to simulate the boost phase of the missile's flight. A study was made of the various impactors then available. There wasn't a great deal of information on the subject but the concensus seemed to favor the drop-test type of impactor.

The general scheme was to accelerate the specimen to the required velocity by means of a free fall and then decelerate the specimen with some energy absorbing system. A sand drop tester is an example of this type of impactor. Another drop tester uses a system in which steel blades shear through lead slugs to provide the deceleration force. These systems occupy a lot of space and did not satisfy our requirements for performance and repeatability. Therefore, we concentrated our design efforts on a pneumatic impactor.

The pneumatic impactors that were designed are easy to operate; require a minimum floor space; give results that are repeatable with a high degree of accuracy; are adjustable to give a range of deceleration levels, rise time to the deceleration level, and hold time at the deceleration level; and allow hydraulic and electrical connections to the missile during an impact so the performance of the missile can be checked during the test.

Two impact testers were designed and built. The first was designed and tested in 1951. It was the prototype used to prove out the theory and now serves as a missile components impactor. The missile impactor was designed and tested in 1953 with the first missile impact being made in October 1953. Patents were applied for and the patents are pending at the present time. The missile impactor is capable of impacting items weighing as much as 4,000 lb. For the most part I will confine my remarks to the missile impactor. (See Figures 1 and 2.)

The operating principle of the impact tester is relatively simple. Two opposed pistons in a pneumatic cylinder are the primary accelerating and decelerating components of the tester. An accelerating force is provided by compressed air behind the accelerating piston from an accumulator which surrounds the cylinder. The decelerating force is acquired through the decelerating piston which used the combined factors of a compressible column of air between the two pistons and a metered flow of hydraulic fluid within a modified aircraft landing gear shock strut. Acceleration and consequent deceleration forces are thus absorbed at a controlled rate.

Physically, the system consists of the cylinder with the two 22-in diameter pistons in it, a missile carriage for connecting the missile

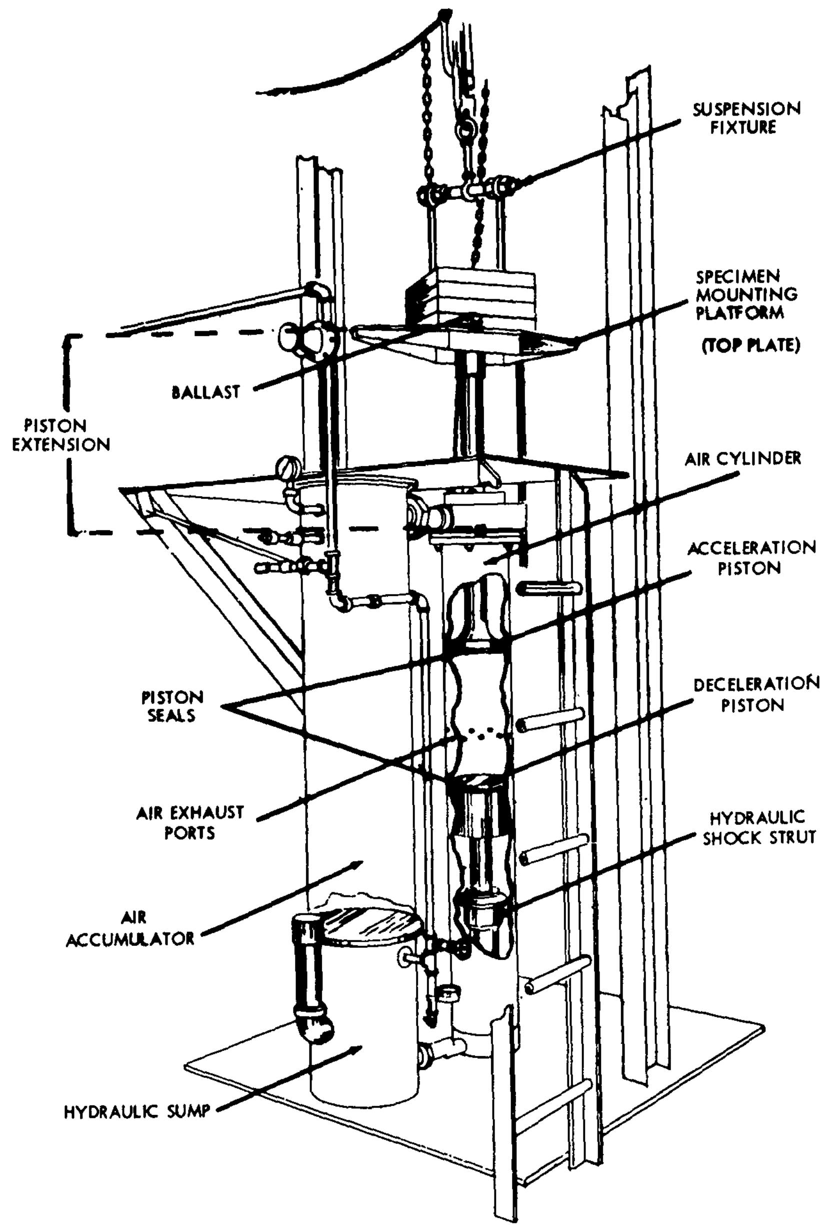


Figure 1 - Design of missile impact tester

to the accelerating piston, and a modified B-47 outrigger shock strut. The missile is mounted on a slide rail by means of a sliding shoe at the missile's center of gravity. The aft end of the missile is clamped to the missile carriage. The slide rail permits seven ft of travel with the first five ft being used to accelerate the missile to the proper velocity and approximately two ft of travel being used to decelerate the missile. The missile carriage and the accelerating piston are connected by a four-in diameter steel shaft. Eight strain gages are mounted on the shaft in a temperature and bending compensating bridge to form a calibrated dynamometer to measure the force on the shaft during an impact. The output of the bridge is amplified and recorded on a Consolidated Electrodynamics Corp. recorder.

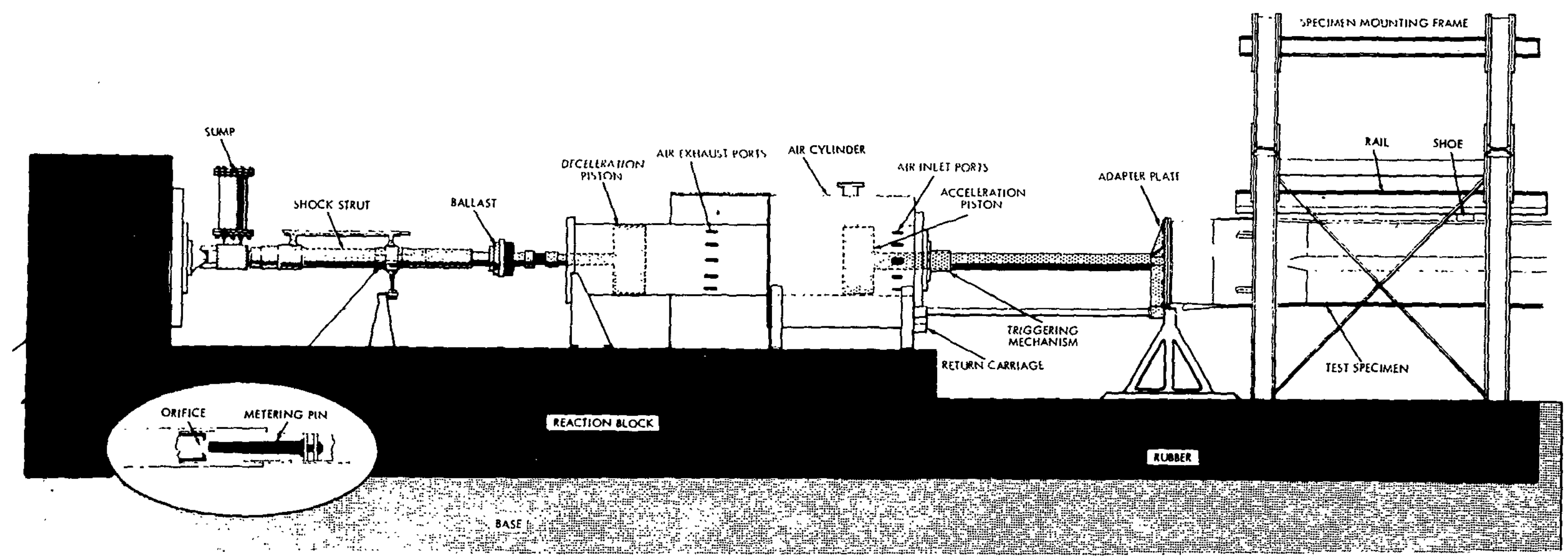


Figure 2 - Missile impact tester

The decelerating piston is connected to the metering pin in the shock strut. The cylinder has inlet ports from the air accumulator tank to allow the expanding air to accelerate the first piston. The exhaust ports in the cylinder are five ft from the starting position of the first piston. These exhaust ports are approximately one ft in front of the second piston. The shock strut contains the sump filled with hydraulic oil, and the orifice which in conjunction with the metering pin determines the rate of energy absorption in the shock strut.

The ultimate shock load is absorbed by a 90-ton concrete block mounted on a 1/2-inch thick rubber pad and contained in a 140-ton concrete foundation. The foundation of the unit is isolated from the foundation of the building in which it is housed.

The easiest way to describe the operation of the impactor is to outline the procedure followed in making an impact on a missile. Before an impact is made, the free travel of the missile carriage is checked by releasing the pneumatic trigger mechanism which locks the accelerating piston and slowly raising the air pressure in the air accumulator until the missile carriage moves. The missile moves only a short distance as the air pressure is kept below 10 psi. This check is made to determine if there is any binding anywhere in the system and is a measure of the starting friction. The carriage usually moves with 2 to 5 psi air pressure. After this friction check is made the carriage is returned to the starting position; the trigger mechanism is set; the shock strut is checked to make sure the decelerating piston is in the forward position and the sump has the proper oil level; and the air pressure raised in the accumulator. The impactor is now ready for an impact.

The impact cycle begins when the hydraulically operated trigger mechanism releases the accelerating piston. The high-pressure air behind the piston and in the accumulator expands and drives the piston and missile carriage forward. During this pre-impact phase the missile is accelerated at approximately 5 g's to a velocity of 30 fps measured at the time the piston crosses the exahust ports. After the piston crosses the exhaust ports the air trapped between the two pistons is compressed which starts to decelerate the first piston and accelerate the second piston. The movement of the decelerating piston is resisted by the action of the shock strut in forcing oil through an orifice. The deceleration force set up in the shock strut is a function of the metering pin profile and the velocity of the second piston. The hydraulic force must decelerate the missile at the specified rate and hold the deceleration for a period of time.

Briefly, the impact can be separated into three phases: the first is a velocity acquiring stage; the second is the deceleration or "rise time" phase which is started when the first piston covers the exhaust ports and starts to compress the air between the two pistons; and the third is the "hold time" phase which is started when the tip of the metering pin enters the orifice. Before we study the design of the impactor it will be helpful to discuss each phase of the impact in more detail.

The first, or velocity acquiring stage, doesn't present any real problem as there is ample room to accelerate the missile to the desired velocity. With a total missile and carriage weight of 3,000 lb an accumulated air pressure of 35 psi will accelerate the system to a velocity of 30 fps as the first piston crosses the exhaust ports. This velocity is adequate for a 30 g impact with a 0.025 second hold time. The second, or "rise time," phase presents more of a problem as we have a moving system which must transfer some of its kinetic energy to the second piston to accelerate it to a common velocity in 0.020 seconds and at the same time the second piston must supply a resistive force to decelerate the missile at the specified rate. The trapped air between the two pistons acts as a spring to transfer the kinetic energy from the missile and first piston to the second piston and shock strut.

The shock strut must provide the resistive force to decelerate the missile at the proper rate. The spring constant of the air column is a function of the air column length, and the hydraulic force is a function of the orifice area. The air column length and the distance from the tip of the metering pin to the orifice were made adjustable to permit some tailoring of the deceleration curve. With a short air column length and a short metering pin to orifice distance, the rise time is short and the G value high but the kinetic energy of the system is dissipated too rapidly so the hold time is very short. There is a strong possibility also that the impact will show an oscillation at the end of the rise time phase which indicates that the two pistons did not have the same velocity at the end of the rise time phase. With a long air column length and a long distance from the metering pin tip to the orifice, the rise time is long and the G value low but the kinetic energy is not dissipated so the hold time is long.

The third, or hold time, phase is tied in with the second phase rather closely. The third phase starts when the tip of the metering pin enters the orifice. The two pistons are traveling at the same velocity so the resistive force of the hydraulic oil being forced through the annular opening between the metering pin and the orifice must decelerate the moving masses at a constant rate to give the required hold time. The hydraulic force is proportional to the velocity squared and the velocity must be decreasing at a constant rate to maintain the deceleration. The problem is to design a metering pin with a profile which changes the annular opening between the metering pin and the orifice at a rate which will maintain a resistive force to obtain this constant deceleration.

Now that we have looked at each phase of the impactor briefly the next step is to study the design of the impactor and its performance characteristics. Our impactors were designed by P. E. O'Neill, W. J. Krause and R. L. Reed of Bendix Aviation Corporation and a good share of the information I'm supplying today is taken from their reports. I don't intend to go into any detail but merely want to outline their solution to the problem of designing an impactor. The positive acceleration phase which brings the missile carriage and the accelerating piston to the proper velocity was calculated by allowing a given volume of compressed air to expand behind the first piston thus accelerating the piston and missile carriage. The air expansion was assumed to be adiabatic.

For a typical missile impact 35 psi air pressure is sufficient to accelerate the missile to a velocity of 30 fps as the first piston crosses the exhaust ports and starts the deceleration phase. A free-body diagram of the system shows that essentially we have two masses with interrelated forces acting upon them. Assuming our analysis starts when the first piston crosses the exhaust ports, the free-body diagram of the first mass shows a single force acting to decelerate the mass. This force is the resultant of the compression of the air between the two pistons. The second mass has this same force acting upon it but in a direction which accelerates the mass. In addition to the force of the compressed air column the second mass has a resistive force generated in the shock strut as the movement of the second piston forces hydraulic oil through the metering orifice. The shock strut force resists the acceleration of the second mass.

The differential equations of motions for the two masses have the force of the compressed air of the air column expressed in terms of the relative motion of the two pistons. The air compression was assumed to follow an adiabatic compression. The metering pin was not inside the orifice during the "rise time" phase so the hydraulic force was neglected for the calculation of this phase. The differential equations of motion for the two masses were combined in a single equation in terms of the relative displacement between the pistons. The equation was integrated. The constant of integration was found from the condition that when the relative displacement equaled the original air column length, the first derivation of the relative displacement equaled the velocity of the accelerating piston as it crossed the exhaust ports. The result of this integration didn't give the velocities of the two masses but gave the relationship between the difference of the two velocities.

Equations for the final velocity and the deceleration of the two masses were derived by using a work-energy relationship and the boundary conditions. The boundary conditions were that the final air column length must provide a compressive force to decelerate the first mass and also that the velocity of the two masses be the same and equal to the velocity required for the deceleration-time specification. These equations were expressed in terms of mass one, mass two, the original air column length, and the velocity of mass one as it crossed the exhaust ports. The "hold time" phase was calculated from the differential equations of motion of the two masses with the hydraulic force acting on the second mass. The deceleration of the two masses was assumed equal at the end of the "rise time" phase. The resulting equation was in terms of the velocity of the second mass and the system parameters, one of which is the annular opening between the metering pin and the orifice. The metering pin diameter was calculated for each increment of pin length using data from a curve of velocity vs. displacement for the second mass. The velocity and displacement vs. time curves were obtained by graphical integration of the acceleration expressions. The calculated deceleration-time curve was of the desired form and the means of modifying this form were indicated from the calculations. The design of the impactor allowed for changes in the air column length, the distance from the tip of the metering pin to the orifice, the weight of the second mass, and the metering pin profile. The metering pin profile change is time consuming but the other changes are easily made.

Now that we've discussed the operation and design of the impactor, we will look at the performance of the impactor and see what it will do.

A series of drawings have been prepared to show the effect of varying the system parameters. The curves shown are recorded from the strain gage shaft dynamometer readings. Figures 1, 2, and 3 show impacts that were made on the missile impactor:

Curves 1 and 2 (Figure 3) provide a comparison of two impacts under the same test conditions.

Curves 18, 19, and 20 (Figure 7) show the effect of varying the air column length.

Curves 21, 22, and 23 (Figure 8) show the effect of varying the metering pin to orifice distance.

Summarizing, we have looked at the operation, design, and performance characteristics

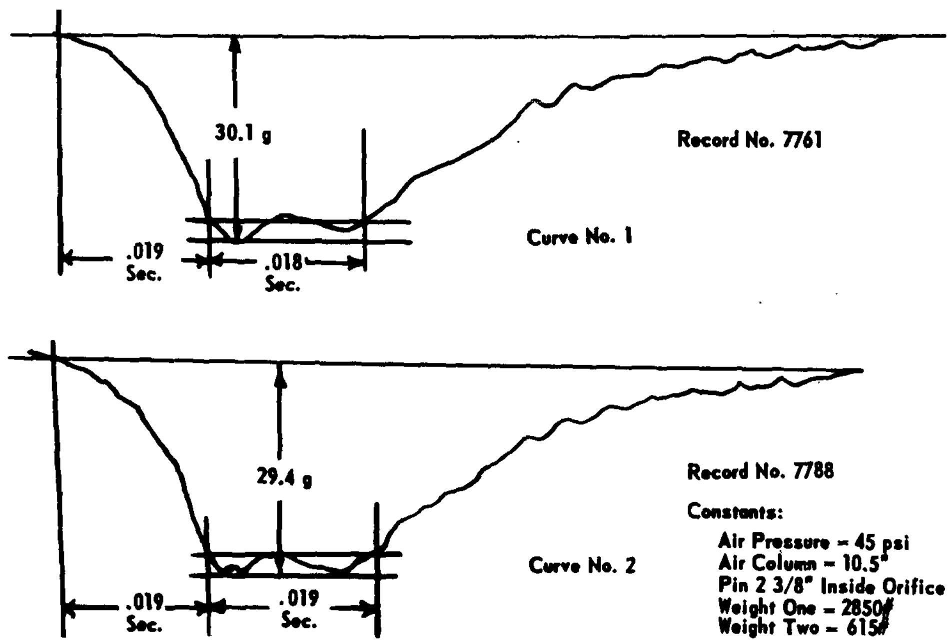


Figure 3 - Comparison of two impacts under same conditions

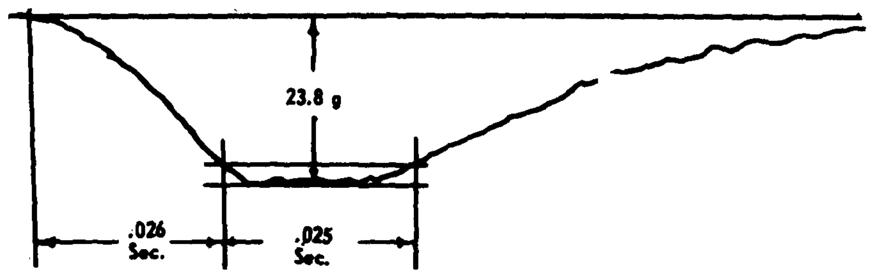
Curves 3, 4, and 5 (Figure 4) show the effect of varying the accumulator air pressure from 45 to 65 psi.

The remaining figures show impacts that were made on the small component impactor:

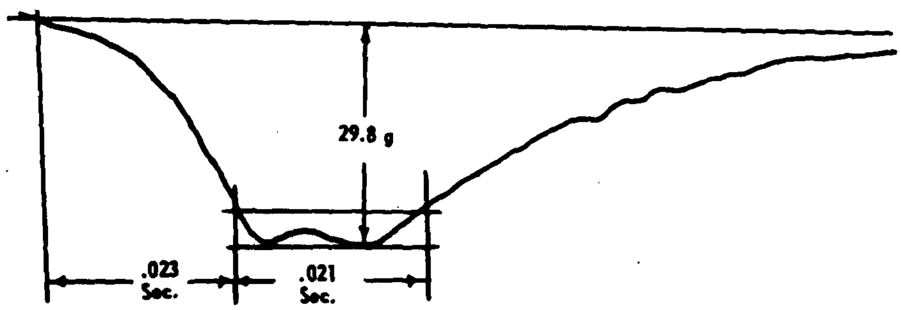
Curves 9, 10, and 11 (Figure 5) show the effect of varying the accumulator air pressure.

Curves 12, 13, and 14 (Figure 6) show the effect of varying the accumulator air pressure. These curves are the same as 9, 10, and 11 except the metering pin to orifice distance is 1.14 in instead of 0.14 in.

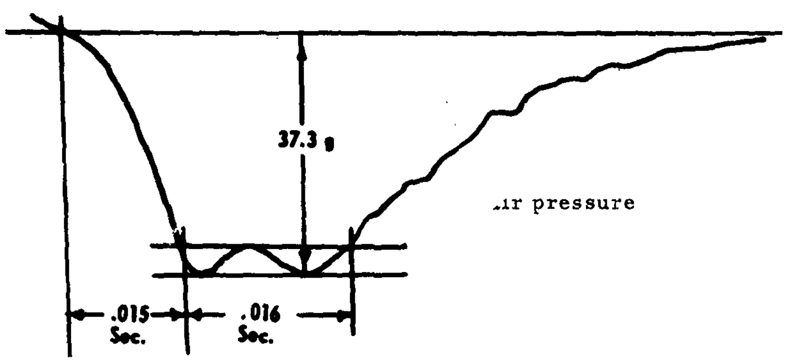
of the missile and component impactors used at the Bendix Missile Facility to simulate the start of the boost phase of the Talos missile. The deceleration obtainable with the small impactor, with 250 lb the maximum weight of test specimen, is 40 g's with a "rise time" at 25 g's of 0.015 sec and a "hold time" at 25 g's of 0.028 sec. The deceleration obtainable with the large impactor, with 3,000 lb maximum weight of test specimen, is 100 g's with a "rise time" of 0.020 sec at 25 g's and a "hold time" of 0.025 sec at 25 g's. The impacts are repeatable within ±5 percent in maximum g value, rise time, and hold time and each impact is accomplished with a minimum of time and almost no expenditure of material.



Curve No. 3, Record No. 7850, 45 psi



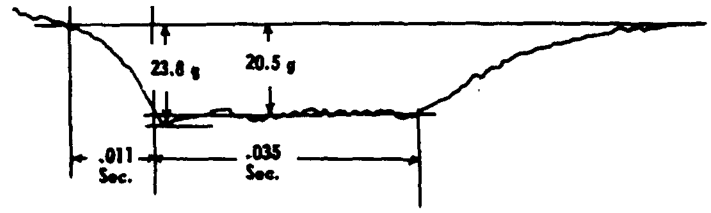
Curve No. 4, Record No. 7852, 55 psi



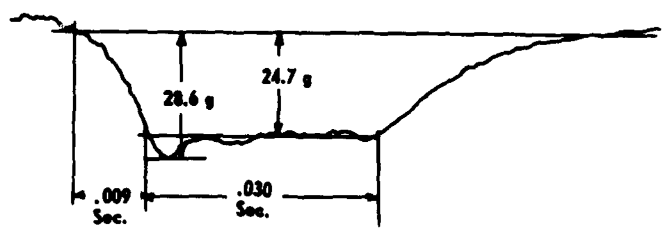
air pressure

Curve No. 5, Record No. 7854, 65 psi

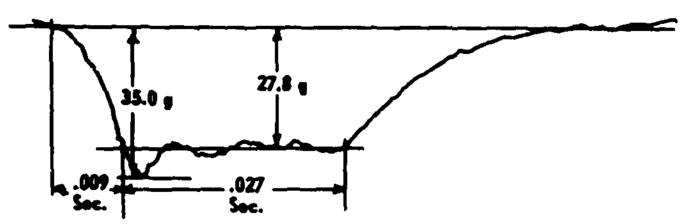
Figure 4 - Effect of varying air pressure. Air column = 13.5 in., pin 0.25 in. outside orifice, weight one = 2850 lb, weight two = 615 lb.



Curve No. 9, Record No. 3501, 60 psi

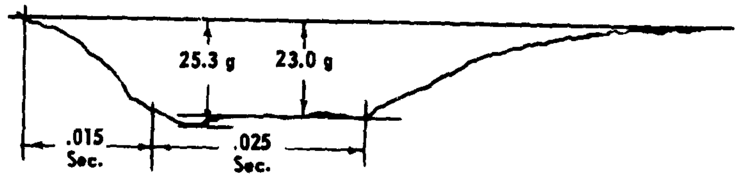


Curve No. 10, Record No. 3505, 70 psi

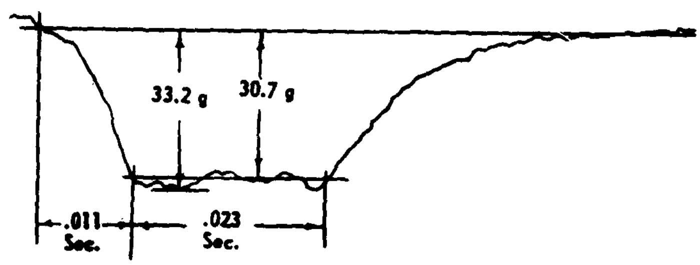


Curve No. 11, Record No. 3512, 80 psi

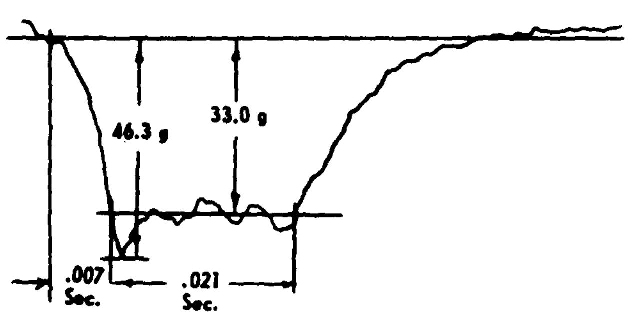
Figure 5 - Test runs with varying accumulator pressures. M_1 = 304 lb, piston ext. = 39 in., air column = 5 in., pin = 1-1/4 in. outside, 1-in spacer installed.



Curve No. 12, Record No. 1146, 50 psi

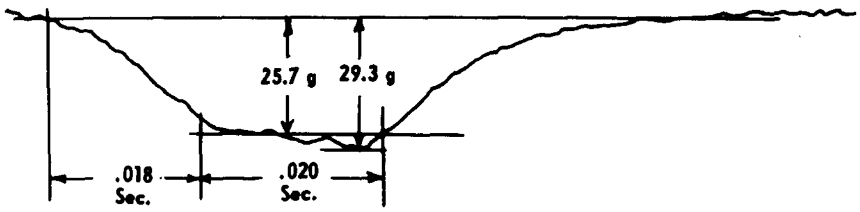


Curve No. 13, Record No. 1151, 70 psi

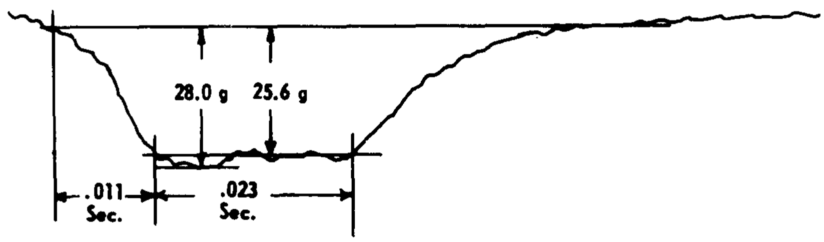


Curve No. 14, Record No. 1158, 90 psi

Figure 6 - Test runs with varying accumulator pressures. M_1 = 304 lb, piston ext., = 39 in., air column = 5 in., pin = 0.14 in. outside orifice, 2-in spacer installed.

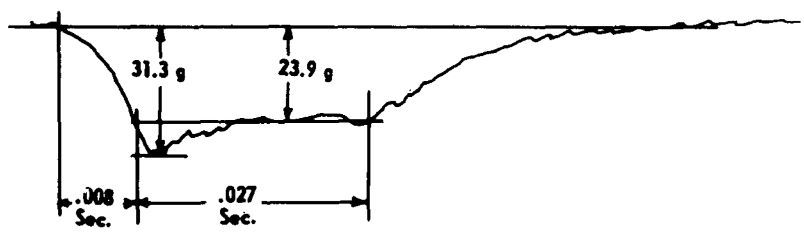


Curve No. 18, 7 in. air column, Record No. 1374, no spacer



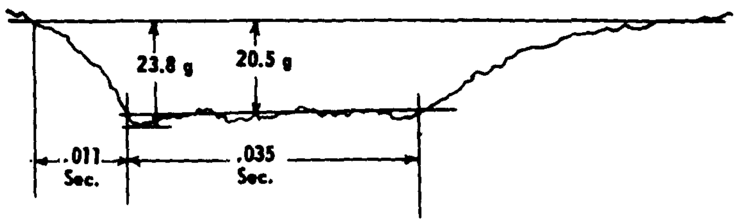
Curve No. 19, 5 in. air column, Record No. 1149, 2-in. spacer

Figure 7 - Test runs with varying air columns

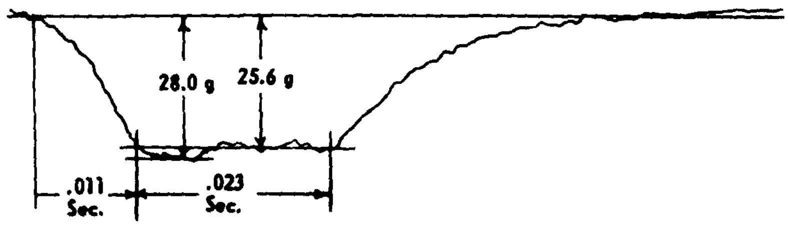


Curve No. 20, 4 in. air column, Record No. 1328, 3-in. spacer

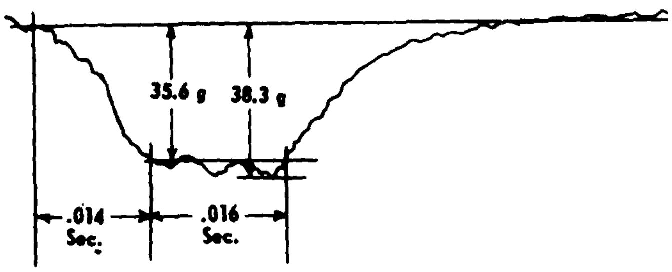
Figure 7 - Test runs with varying air columns. M_1 = 304 lb, piston ext. = 39 in., pin 0.14 in. outside orifice, accum. pressure = 60 psi.



Curve No. 21, Record No. 3501, pin 1.14 in. outside orifice, 1-in. spacer



Curve No. 22, Record No. 1149, pin 0.14 in. outside orifice, 2-in. spacer



Curve No. 23, Record No. 1325, pin 0.86 in. inside orifice, 3-in. spacer

Figure 8 - Test runs with varying pin positions. M_1 = 304 lb, piston ext. = 39 in., air column = 5 in., accum. pressure = 60 psi.

DISCUSSION

Rice, Goodyear Aircraft: I notice you have 30 g on there for about 13 milliseconds. Is that representative or just an arbitrarily g load in the boost phase?

Sanders: I think that the hold time is somewhat short but the specifications actually call for a 30 millisecond hold time. I believe that the boost phase covers a longer period than that, but we are just trying to achieve the start of it. Our knowledge of the science hasn't progressed to the point where we can get a longer hold time.

* * *

DEVELOPMENT OF VIBRATORS FOR IMPROVED SIMULATION OF REAL VIBRATION ENVIRONMENTS

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Alexandria, Va.

Two vibrators for testing small electronic components are discussed. One is a magnetostrictive unit with a flat response (20 percent) between 1 kc and 10 kc. The second is a liquid jet vibrator producing a nearly uniform acceleration spectral density between 1 kc and 10 kc.

INTRODUCTION

The Need for Improved Simulation

Laboratory vibration tests have been proven to be of economic utility in simulating the vibration problems involved in low-speed transportation, and with the provision of suitable means for simulating the mechanical environment produced by high-speed vehicles it becomes increasingly imperative to keep pace with our ability to make things travel further and faster with more complex payloads.

The solution of any engineering problem requires either a background of data directly applicable to the problem at hand or a logical means of interpolating or extrapolating known data into the data required for the solution. Applying this criterion to a vibration test, we can see that logical interpolation or extrapolation of, say, a prediction of failure requires a knowledge of the resonant frequencies and accompanying quality factors for the structure under test. Most components subjected to vibration tests are of such complex mechanical construction that knowledge of the resonances can be obtained only empirically, and thus one is forced to accept the necessity of providing empirical data covering the entire range of the application.

In recent years, much effort has been applied to the determination of the amplitude and frequency ranges of the vibrations encountered by the guidance, control and instrumentation components (primarily electronic) used in rockets, and in jet-powered missiles and aircraft. These continuing studies have resulted in a better, though as yet incomplete, definition of the mechanical environment, and it is generally accepted that this environment consists of forces more or less randomly distributed in space and time. By recording these forces over a sufficient period of time, the environment may be transformed into an equivalent spectral density of forces as a function of frequency. Two conclusions have so far been drawn:

1. The spectral density of the forces is continuous, i.e., the forces are not those which might be produced by the sum of a finite number of vibration sources, each with some associated discrete frequency, but is the type of distribution generally called "noise";

2. The upper frequency limit beyond which such forces are unimportant has not been defined.

Simulation of this environment by vibrators driven at single (variable) frequencies is, at

best, a tedious task. As is pointed out in References 5 and 7, such simulation requires in principle a complete knowledge of the response of the item under test before a logical choice can be made of the amplitude variation and rate of sweep used in a sweep-frequency test or of the amplitude and frequency of a series of single-frequency tests which would subject all parts of the system tested to forces comparable to those encountered in use. The vibrator needed is thus one which will produce a complex force distribution closely reproducing or enveloping that found in service.

The Development Program

During negotiation of the contract* under which the work described herein was performed, and in subsequent discussions of the direction of effort, the following objective was defined:

> The development of a vibrator or vibrators capable of producing a constant (±20 percent) acceleration spectral density (rms acceleration in a frequency band one cycle wide) with a total rms acceleration of 100 times the acceleration of gravity in the frequency band 1,000 to 10,000 cps, with a useful load of at least 10 grams.

This objective may be obtained by the use of a driving mechanism which by its very nature produces such a noise spectrum, or by driving an electromechanical transducer, having a flat frequency response characteristic, with an electronic noise generator.

A considerable amount of development effort had been applied, at the Diamond Ordnance Fuze Laboratories and elsewhere, to approach this same objective using an electrodynamic vibrator. Although some thought was given to the possibility of further improvement of such vibrators, our principal efforts were directed to the theoretical and experimental evaluation of techniques on which little, if any, development effort had previously been applied with the above stated requirements in mind.

After a brief survey of the developments then being pursued by others, it was decided to divide our efforts and investigate:

1. The feasibility of making a magnetostrictive transducer with an essentially flat frequency characteristic between 1,000 and 10,000 cps, and
2. The accelerations which can be produced by a fluid jet impinging on the mass to be vibrated.

The results of these efforts are detailed in other sections of this paper.

Measurement of Acceleration

The provision of tools for the measurement of accelerations at frequencies above 2,000 cps has been given increasing attention in recent years (References 1, 2, and 10). At the present time there are on the market several commercial accelerometers which could conceivably be used at frequencies as high as 30 kc, but their use requires the extrapolation of calibration data obtained in the neighborhood of 1,200 cps. Such an extrapolation is at best undesirable, and the provision of accurate means of calibration of such instruments over the entire frequency range in which measurements are to be made remains the most important single problem facing the users of accelerometers.

We have not provided a solution for this problem, but our confidence in the measurements made is somewhat increased by the fact that we have had available an accelerometer, the Model T-3, developed at the National Bureau of Standards (Reference 10), that has been used for some time for measurements at frequencies up to 10 kc with no evidence of substantial deviations from a flat sensitivity vs frequency characteristic.

The data presented in this paper have been reduced to values of acceleration using the calibration provided by the Diamond Ordnance Fuze Laboratories, which calibration is accomplished using an electrodynamic vibrator (Reference 11). The sensitivity is observed at an accurately known acceleration level at a low frequency (usually 60 cps) and then the sensitivity vs frequency characteristic is observed on a panoramic wave analyzer at frequencies up to 5 kc. Experience with the electrodynamic vibrator has shown that its output is quite flat in this frequency range, and in the absence of any significant variations in the accelerometer output it was assumed that the low frequency calibration could be extended to 10 kc.

Some hope for better resolution of the calibration problem is provided by work presently being done by the Office of Basic Instrumentation, NBS (Reference 1). An optical interferometric method of measuring accelerations greater than 0.50 g at 1,000 cps, or 50 g at 10 kc

*Contract No. DAI-49-186-502-ORD(P)-210.

has been developed, and we are presently planning to accept the opportunity presented by the cooperation of NBS and DOFL to obtain an improved calibration of the accelerometers we have used and to determine the feasibility of using the magnetostrictive unit as a secondary calibration standard.

An additional method of calibration which should prove useful is a reciprocity calibration (References 3, 12, and 13). We are here, however, approaching the frequency range where the use of lumped parameters to describe the coupling impedance needed in one of the measurements for a reciprocity calibration is no longer valid, and wave effects in this coupling must be considered.

Attachment of the accelerometer to our vibrators was accomplished by machining the outside of the accelerometer housing to a taper of 1/4 in per ft, and providing a matching taper in a mounting hole in the solid blocks used as the load.

The necessary impedance transfer from the high-electrical impedance of the accelerometer to the relatively low electrical impedance of the measuring equipment used was accomplished using a cathode follower with approximately 50 megohm input impedance.

A WIDE BAND MAGNETOSTRICTIVE VIBRATOR

The possibility of developing a transducer with a flat frequency response over a relatively wide band in the audio frequency range was indicated by the work of Rabinow and Apstein (Reference 8) at the National Bureau of Standards.

The sharp resonant peaks characteristic of the frequency response of most electromechanical transducers may be visualized as being caused by the reflection of elastic-wave energy at the boundaries of the transducer in such a manner as to reinforce the energy subsequently supplied by the driving mechanism. A nearly uniform response may be obtained by restricting the use of the transducer to frequencies well below that at which the first such resonance occurs, but this restriction generally results in a severe limitation of the total amount of power which may be transformed from electrical to mechanical form. Rabinow and Apstein showed that much larger amounts of power could be handled and the efficiency of the energy transformation maintained substantially independent of frequency over a wide frequency range by the use of means to eliminate unwanted reflections of energy at the transducer boundaries, and by controlled reinforcement of the energy transformation.

Our efforts were thus applied to the evaluation of this method of approach to the problem at hand. A magnetostrictive transducer was chosen for study, primarily because of the convenience with which the most common magnetostrictive material, Grade A nickel, may be worked, and because of the convenient electrical impedance values of such a completed transducer. The results of this investigation may be applied to other types of transducers, and it is conceivable that use of other electromechanical effects might yield improved performance or allow a reduction in the size of the transducer for the same output.

The active element of the transducer was in all cases a single piece of nickel tubing, and the problem of absorbing the elastic wave energy at one end of this tube is analogous to that of the termination of an electrical transmission line. The electrical line may be terminated with a resistor to absorb all of the incident electrical energy, but no such simple arrangement is possible in the mechanical case since no one has yet invented a pure mechanical "resistor."

Rabinow and Apstein approximated such a mechanical resistor by the use of the friction between the transducer element and one or more masses lightly clamped or held by magnetic attraction, but this approximation was not sufficiently good for the objective we wished to achieve. Our best success was obtained by terminating the transducer element with a "lossy" mechanical transmission line, i.e., a piece of lead tubing, whose length could then be chosen to give any desired degree of attenuation of the wave energy being transmitted. The use of friction was not completely discarded, however, and an additional loss was obtained by winding the lead tubing with wire solder.

Experimental Apparatus

The form of the vibrator is shown in Figure 1. The active element of the transducer consists of a section of Grade A nickel tubing, machined to 0.650-in O.D., 0.020-in wall thickness, 47 in long. Both the static (polarizing) magnetic field and the driving field are produced from a single layer helical coil of #18 magnet wire (approximately 600 total turns) wound loosely over the nickel tube. At one end is brazed a 5/8-in diameter by 3/4-in long rod of nonmagnetic stainless

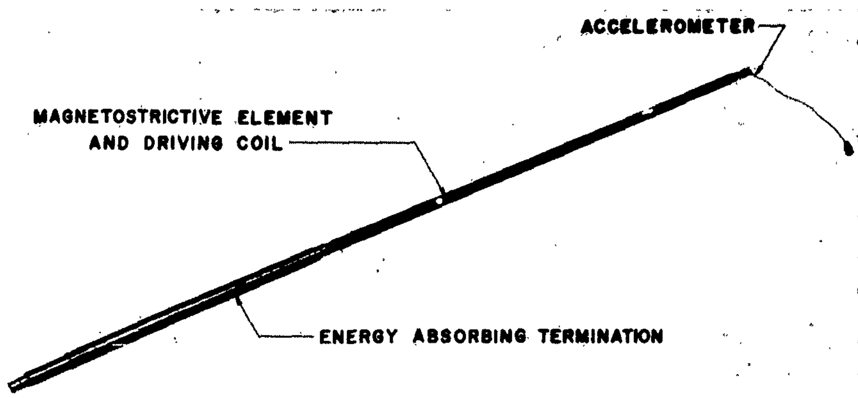

Figure 1 - The wide-band magnetostrictive vibrator

steel provided with a mounting hole for the accelerometer. The other end of the active element is soldered to a 24-in length of lead tubing of 0.650-in O.D., 1/32-in wall thickness, fabricated from lead sheet. These dimensions are chosen to provide the same mechanical impedance per unit length as that of the nickel tube. Additional power dissipation, beyond that provided by the internal damping of the lead, is produced by external windings of 1/16-in diameter wire solder in light contact with the lead tube.

An over-all schematic of the vibrator and electrical and electronic equipment needed to produce and measure the vibrations is shown in Figure 2. The driving signal may be supplied

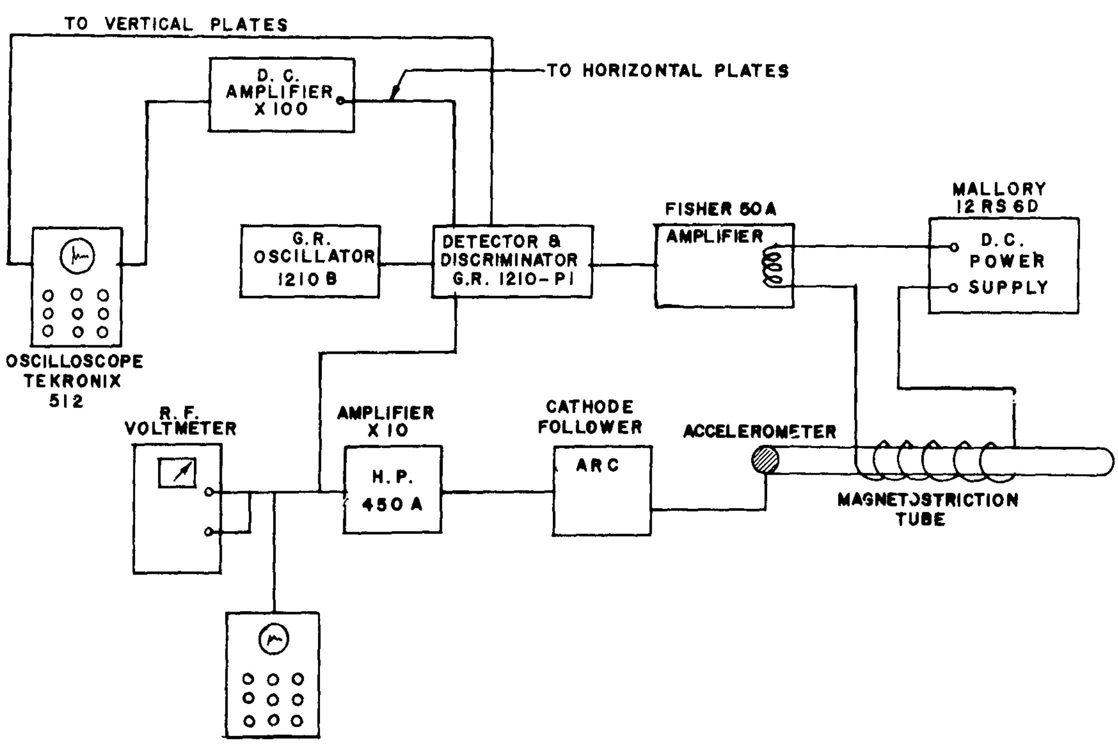

Figure 2 - Magnetostriction vibrator circuitry

by an oscillator, a sweep-frequency oscillator or a random noise generator. This driving signal is amplified by a 30-watt power amplifier, and fed to the driving coil in series with the output of the dc power supply. A direct current of one ampere provided the necessary polarizing flux.

NBS Type T-3 accelerometers and the cathode follower as mentioned in Part I were used for the measurement of accelerations. The detector-discriminator circuit was used in conjunction with the sweep-frequency oscillator to display the frequency response curve of the vibrator on a long persistence screen of a cathode ray tube for ease of observation of the effects of variation of some of the vibrator parameters. The waveform of the accelerometer output was monitored on another cathode ray tube. For more precise measurements, an audio-frequency oscillator was used as the source of the driving signal, and measurements of electrical input and accelerometer output being made with a vacuum tube voltmeter, the output waveform being monitored as before.

Experimental Results

The performance of the vibrator with a constant ac voltage of 30 volts rms supplied to the driving coil is shown in Figure 3. The total load driven, consisting of the stainless steel accelerometer holder and the accelerometer, was approximately 25 grams.

It may be seen that the performance very nearly meets the desired specifications. The actual output is flat within 25 percent from 1,500 to 10,000 cps, the average output level being approximately 70 g rms. Some distortion of the waveform of the accelerometer output was noted at frequencies below 2,500 cps.

Subsequent attempts to improve the vibrator performance by reducing the wall thickness of the nickel tube (thus reducing eddy current losses) and annealing the tube after the machining operations resulted in performance as shown in Figure 4. (The total driven load was also reduced to approximately 15 grams.) A substantial deviation from uniform response is noted at the higher output level.

Discussion and Conclusions

An elementary theoretical analysis of the performance of the vibrator described in detail above is contained in Appendix A, and results of this analysis are compared with the measured performance in Figure 3. The differences

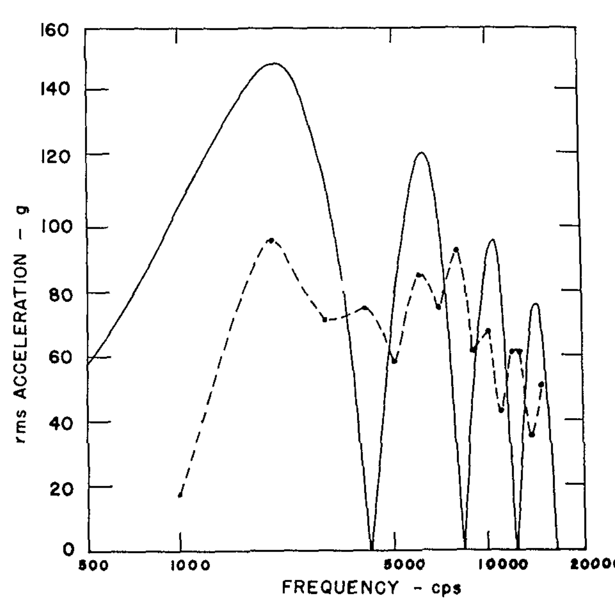

Figure 3 - Constant voltage response of magnetostrictive transducer: theoretical response of loss less transducer (see Appendix A) measured response

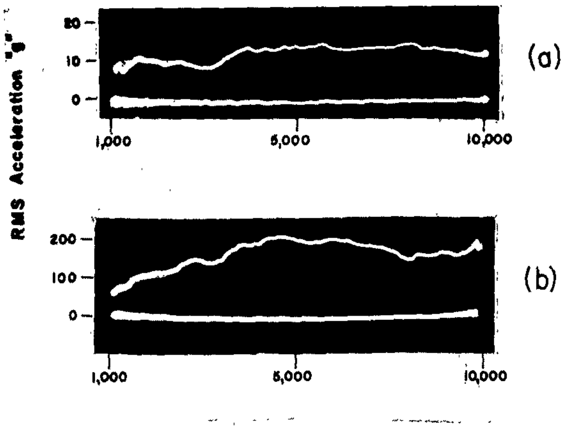

Figure 4 - Acceleration response

between theory and practice may be attributed to the effects of the finite resistance of the driving coil, eddy current losses in the nickel tube, imperfections in energy transfer between the nickel rod and lead termination, and the finite length of the termination itself, all of which were neglected in the theoretical analysis. It is seen that some idea of the output to be expected from the vibrator, and the variation in this output over wide frequency ranges may be obtained from simple theory, but that a prediction of the details of the performance, including specifically the degree to which a uniform response is approached, depends on damping effects and other imperfections which are not subject to simple analysis.

Preliminary tests of the vibrator, using a random-noise generator to produce the driving signal, have verified the practicability of such excitation, and have substantially confirmed the frequency response characteristic observed in the sweep-frequency tests.

The techniques described here have not yielded a solution completely satisfying the objective defined at the inception of the development, but have provided a device capable of producing 150 g accelerations of a 15 to 20 gram mass at frequencies from 2,000 to 10,000 cps. Variation of the output with constant voltage input is within ±35 percent.

A LIQUID JET VIBRATOR

During the development and in subsequent testing of the electrodynamic vibration generator developed at the Diamond Ordnance Fuze Laboratories (Reference 11), it was observed that the sound radiated by this vibrator, when driven by a random-noise source, was quite similar to that produced by the high-velocity flow of water or other fluids. Thus at the inception of the work done by the Atlantic Research Corporation under DOFL sponsorship, it was agreed that an investigation of the possibility of coupling the turbulent energy of a fluid directly to the load to be vibrated would be undertaken. A prime advantage of such a system would be complete freedom from electrostatic and electromagnetic fields produced by the driving mechanism, and thus the effects of a purely mechanical environment might be more easily determined.

A theoretical estimate of the magnitude of the fluctuating forces produced upon a flat disc inserted in a turbulent stream led to the conclusion that a total input power of 43 kw would be required to provide a random excitation of 100 g rms for a total mass of 30 grams. This calculation was made by applying present knowledge of the turbulence generated by wire grids in low-speed air streams (such that the air may be considered incompressible) to the flow of water, and assuming that the coupling can be approximated by the use of a drag coefficient, thus neglecting the effects of the relative size of the turbulent eddies and the body being driven. This power requirement appeared somewhat impractical, and led to the consideration of other methods of coupling to the random energy in the stream.

The vibrator designed for preliminary testing took the form of a free liquid jet impinging on the load to be driven, this load being supported by a flexible diaphragm. By the use of this configuration, we hope to be able to avoid the damping which would occur if the vibrating body were completely surrounded by the liquid, and to take advantage of the possible increase in the fluctuating energy component as a result of disintegration of the jet. At that time it appeared impractical to make any attempt to predict the performance of such a system, and a completely empirical approach was adopted. Subsequent considerations of the probable mechanism which produces the desired vibration has led to a simplified analysis which appears to have some utility in predicting the order of magnitude of the accelerations which may be produced by this vibrator. This analysis is included in the discussion below.

Experimental Apparatus

A schematic drawing of the nozzle, load, and diaphragm arrangement used is shown in

Figure 5. Water was used as the driving fluid, and pressures up to 300 psig upstream of the nozzle were produced by a gear pump rated at one gallon per minute at 200 psig. Subsequent modifications to this unit allowed the use of nozzle-to-diaphragm distances up to 23 in.

would be introduced by this spurious passband of the analyzer required the use of the low-pass filter. Since our immediate attention was focused on the 1 to 10 kc band, this expedient did not interfere with our aims. If, however, we wish to extend our interest to frequencies of 30 kc,

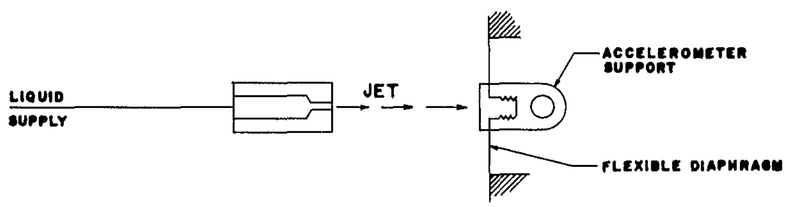

Figure 5 - Schematic of liquid jet vibrator

An extrapolation of the data obtained with this unit led to the procurement of a larger pump rated at 5 gal/min at 1,000 psig. This pump is lubricated by the fluid being pumped. Several unsuccessful attempts to obtain a significant output using the recommended hydraulic oil were made; then the working fluid was changed to a soluble oil-water mixture. This mixture has given some improvement in output, but has resulted in the generation of large amounts of foam, and it has been thus far impossible to obtain any significant results at pressures higher than those which can be produced by the original water pump. The use of a silicone anti-foam emulsion has had no significant effect on the foaming properties of the mixture.

An NBS Model T-3 accelerometer (Reference 10) was mounted in an aluminum block which also was the target for the jet. The output of the accelerometer was fed to a cathode follower through a United Transformer Company LMI-10,000 low-pass filter and to a Hewlett Packard Model 300A Wave Analyzer.

In calibrating the electronic system, it was found that the Wave Analyzer will pass a 20 kc signal independently of its frequency setting. At the same time, the lowest resonant frequency of the accelerometer was checked (using a preliminary model of the magnetostrictive vibrator previously described) and found to be about 21 kc.* Elimination of the error which

*Somewhat below the value reported by Rosenberg (Reference 10), the difference most probably being caused by thinning of the bell-like end when the taper was machined.

as suggested by Kaufman (Reference 5), considerable effort will be needed to provide accelerometers and associated instrumentation to produce accurate data.

Three 2,000-microfarad dry electrolytic condensers were connected in parallel with the output meter of the Wave Analyzer to increase the time over which the output was averaged, and thus reduce the variations in the meter reading.

Data Reduction

The readings of the Hewlett-Packard Wave Analyzer are corrected to yield the rms accelerometer output voltage for a frequency band one-cycle wide. This correction consists of several factors as follows:

1. The voltage gain of the entire measuring system,

2. The bandpass characteristics of the wave analyzer,

3. The reduction of meter readings to rms values for the nonsinusoidal signal.

Correction No. 1 is easily obtained by applying a known single frequency signal at the cathode follower input and measuring the output as this frequency is varied over the range of interest.

Correction No. 2 has been determined from the bandpass characteristics as published in the instruction book supplied with the wave analyzer. For this correction, we assume that the meter

actually measures an rms value. With this assumption the meter reading V_M is given by

$$V_M^2 = \left[Z_M \int_{f_o-\Delta f}^{f_o+\Delta f} \frac{(K_1 V)^2}{Z} \, df \right].$$

Where Z is the impedance of the wave analyzer, Z_M is the average impedance over the full band width $2\Delta f$, K_1 is the voltage gain of the analyzer for a given center frequency f_o (Figure 2 of the wave analyzer instruction book), and V is the input voltage density in a band one-cycle wide. This equation can be solved simply for V only if we assume that V and Z are constant over the band of integration and $K_1 = 0$ outside this band. For $\Delta f = 145$ cps, we have $K_{1(f_o \pm \Delta f)} = 0.01\, K_{1(f_o)}$, thus for the frequency range in which we are interested, 1 kc to 10 kc, these assumptions should not introduce a significant error. The input voltage density is then:

$$V = \frac{V_M}{\left[\int_{fo-\Delta f}^{fo+\Delta f} K_1^2 \, df \right]^{1/2}}$$

This integration was performed graphically.

An additional correction is given by Equation 9 of Reference 7 as

$$V_{rms} = \left(\frac{4}{\pi}\right)^{1/2} V.$$

This correction is applied to account for the fact that the wave analyzer meter actually measures the average value of the output of a full-wave rectifier and is calibrated to read the rms value of a sine-wave signal.

The measured response of the accelerometer is then applied to determine the accelerations obtained. The presumption that the accelerometer sensitivity is constant and equal to the low-frequency calibration value has been used for the entire range of the data obtained.

Experimental Results

The collection and interpretation of data presents considerable difficulty. The addition of 6,000 microfarad in parallel with the output meter of the Wave Analyzer increased the time constant of the meter to more than one second, but variations in output of ±50 percent in several seconds were still apparent, thus the recording of an average value introduces some subjectivity in the data. In spite of these attempts to determine an average of the output, the data still reveal a considerable scatter.

Because of this scatter, a detailed comparison of individual runs is almost completely uninformative, and few specific conclusions can be drawn from the total data. Figure 6, showing three runs with only slight variations in the controlled variables is indicative of the scatter. The best total output obtained to date is an acceleration of approximately 15 g rms over the band 1,000 to 10,000 cps with a total load of 35 grams.

Discussion and Conclusions

An estimate of the magnitude of the acceleration which may be produced by the liquid jet vibrator under optimum conditions is useful for comparison with the experimental data obtained to date, and to ascribe an upper limit to the performance which may be obtained with any given pump characteristics. Some insight into the phenomena which probably occur may be had by a study of References 6 and 9.

In order to calculate the upper bound for the attainable accelerations, we assume that the initially continuous jet disintegrates into a series of discrete drops which all follow the same path in space, and focus our attention on a single drop. This drop will be moving with the average jet velocity, and we assume that all of its forward momentum is converted, at impact, to an impulse on the load being driven. In order to simplify the calculation, we further assume that the impulsive force rises steeply to a value F_1, remains constant for a time interval Δt_1, and then returns to zero. Then

$$mv = F_1 \, \Delta t_1 = M a_1 \, \Delta t_1,$$

where

m = mass of the drop,

v = average jet velocity,

F_1 = impulsive force,

Δt = time interval during which impulsive force acts,

M = mass of driven load,

a_1 = acceleration of driven load during time Δt_1.

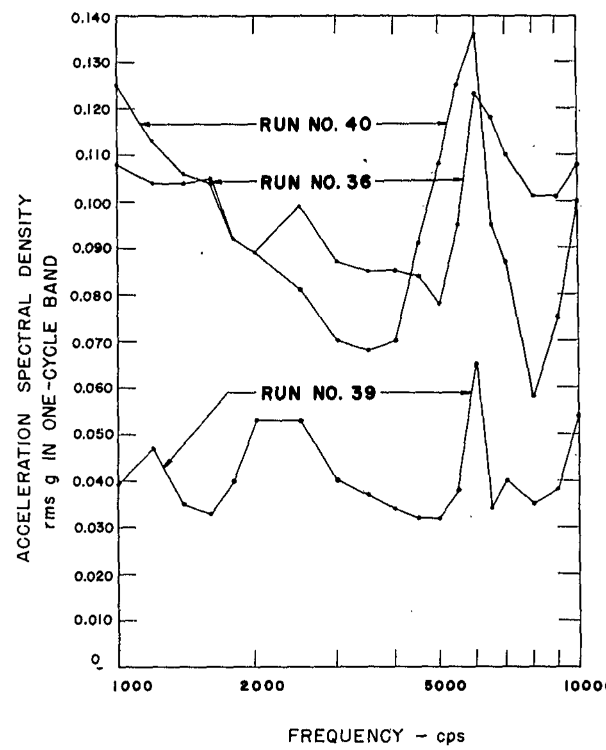

Figure 6 - Comparison of output of liquid jet vibrator for three runs

The mass of the drop is given by

$$m = \rho A \ell$$

where ℓ is the (average) distance between the drop under consideration and its nearest neighbors in the line of drops, ρ is the liquid density, and A is the cross sectional area of the original jet stream.

The total time interval associated with the collision of the drop may be defined as

$$\Delta t_2 = \frac{\ell}{v} ,$$

and combining these equations, we obtain

$$a_1 \frac{\Delta t_1}{\Delta t_2} = \frac{\rho A v^2}{M} .$$

We note that the average value of the acceleration over the time interval Δt_2, $\overline{a}$, is given by

$$\overline{a} = a_1 \frac{\Delta t_1}{\Delta t_2} .$$

We may then calculate the quantity of particular interest, the rms value of the varying component of the acceleration as

$$a_{rms} = \left[\overline{(a - \overline{a})^2}\right]^{1/2} = \frac{\rho A v^2}{M} \left(\frac{\Delta t_2}{\Delta t_1} - 1\right)^{1/2} .$$

Since this value increases without limit as $\Delta t_1/\Delta t_2$ approaches zero, it is necessary to determine if there exists some limiting minimum value for $\Delta t_1/\Delta t_2$. With the simplification already used, we may write

$$\frac{\Delta t_1}{\Delta t_2} = 2r = \frac{2r}{\ell} = \frac{2r}{\lambda}$$

where r is the radius of the drop, here assumed to be spherical, and λ is the wave length, in the jet, of the disturbance which formed the drop being examined. Continuity requires that

$$\frac{4\pi r^3}{3} = A\lambda ,$$

and thus

$$\frac{\Delta t_1}{\Delta t_2} = \left(\frac{6A}{\pi}\right)^{1/3} \frac{1}{\lambda^{2/3}} = (6)^{1/3} \left(\frac{r_1}{\lambda}\right)^{2/3}$$

Rayleigh (Reference 9) shows that the wave length of the fastest growing disturbance in the jet is equal to 9 times the jet radius, but further shows that longer wave lengths may be produced in the presence of sufficiently strong periodic disturbances. Thus we do not have sufficient information to definitely assign an upper limit to λ. Using the wave length for the fastest growing disturbance, $\lambda = 9r_1$, we obtain

$$a_{rms} = 1.17 \left(\frac{\rho A v^2}{M}\right) .$$

This may be considered the most probable upper limit to the accelerations attainable with a single jet in the absence of resonances in the load supporting system.

The assignment of a characteristic wave length to the jet implies that some periodicity occurs in the motion of the stream, and one would thus expect the output to show a tendency toward a peak at a frequency f_o such that

$$f_o = v/\lambda .$$

Examination of individual runs for evidence of such a peak is inconclusive.

The principal conclusions of this experimental program may be drawn from Figure 7, in which the ratio of rms acceleration in the frequency band 1,000 to 10,000 cps to the quantity $\rho A v^2/M$ is plotted as a function of the characteristic frequency of the jet. It appears that the criterion deduced above,

$$a_{rms} \cong 1.2 \frac{\rho A v^2}{M}$$

may be useful in predicting the maximum output for any given jet and load parameters, since the great majority of the data falls below this value.

Two limitations are probably necessary if the output of the vibrator is to approach this maximum value. These are:

1. The characteristic frequency of the jet should be in or near the frequency band of interest,
2. Secondary atomization should be negligible.

This is the disintegration of the jet into drops much smaller than those considered above and is caused by friction with the ambient atmosphere.

It may be easily shown that the breakup of the single drop considered above into η equal drops, effectively increases $\Delta t_1/\Delta t_2$ by a factor $\eta^{2/3}$, even when the energy loss associated with this friction is neglected. This reduces the numerical equation for maximum output, which becomes, for example for $\eta = 2$:

$$a_{rms} = 0.71 \frac{\rho A v^2}{M} .$$

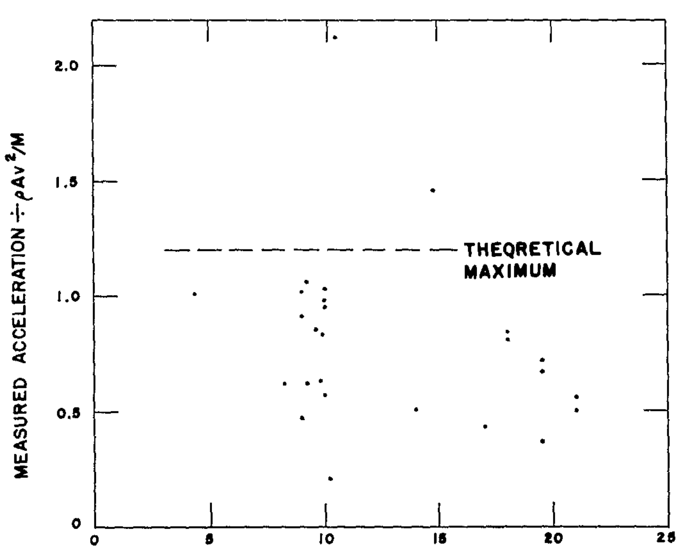

Figure 7 - Liquid jet experimental results plotted to show their relation to the criterion: $a_{rms} \cong 1.2\ \rho A v^2/M$

APPENDIX A

Nomenclature:

$A(\chi)$ = Displacement function

B = Incremental flux density, gauss

C = Velocity of sound, cm/sec

D_1, D_2, D_3 = Constants of integration

E = Modulus of elasticity, dynes/cm^2

e = Base of natural logarithms

f = Frequency, cycles/sec

I = Current through driving coil, abamperes

$i = \sqrt{-1}$

K = Wave number, cm^{-1}

L = Inductance of driving coil, abhenrys

ℓ = Length of magnetostrictive region, cm

m = Concentrated mass load, gm

N = Total turns of driving coil

P = Incremental stress, dynes/cm^2

s = Incremental strain, cm/cm

v = Voltage applied to driving coil, abvolts

$z = \rho c \sigma$, gm/sec

α = Acceleration, cm/sec^2

$\theta = \mathrm{Tan}^{-1}\, z/m\omega$

λ = Magnetostriction coefficient, dynes/gauss-cm^2 (a function of static flux density)

ξ = Displacement from equilibrium, cm

ρ = Mass density, gm/cm^3

σ = Cross-sectional area, cm^2

ω = Circular frequency, rdn/sec

Subscripts:

1 - Medium 1, magnetostrictive

2 - Medium 2, nonmagnetostrictive

The model assumed consists of a length ℓ of magnetostrictive material, rigidly coupled at $\chi = 0$ to a nonmagnetostrictive material, and at $\chi = \ell$ to a concentrated mass, m. The cross-sectional areas of the two materials are chosen such that:

$$\rho_1 C_1 \sigma_1 = \rho_2 C_2 \sigma_2 = z.$$

The damping effect of Region 2 is approximated by restricting the solution in Region 2 to waves propagating in the negative χ direction.

The problem is restricted to one dimension.

The equation of motion for any elemental section is:

$$\frac{\partial P}{\partial \chi} = \rho \frac{\partial^2 \xi}{\partial t^2}.$$

In Region 1,

$$P = -\lambda B + E_1 s \text{ (Reference 14)},$$

and in Region 2,

$$P = E_2 s.$$

Neglecting leakage flux ($\partial B/\partial \chi = 0$) and since:

$$s = \frac{\partial \xi}{\partial \chi},$$

by definition, we may obtain

$$E_1 \frac{\partial^2 \xi}{\partial \chi^2} = \rho_1 \frac{\partial^2 \xi}{\partial t^2}$$

and

$$E_2 \frac{\partial^2 \xi}{\partial \chi^2} = \rho_2 \frac{\partial^2 \xi}{\partial t^2}.$$

Considering solutions of the form,

$$\xi = A(\chi) e^{i\omega t},$$

yields

$$A_1(\chi) = D_1 \operatorname{Cos} K_1\chi + D_2 \operatorname{Sin} K_1\chi ,$$

$$A_2(\chi) = D_3 e^{iK_2\chi} \text{ (Negative } \chi \text{ waves only)}$$

where

$$K_1^2 = \frac{\rho_1\omega^2}{E_1} , K_2^2 = \frac{\rho_2\omega^2}{E_2} .$$

The boundary conditions to be satisfied are

(1) At $\chi = 0 \quad \xi_1 = \xi_2$,

(2) At $\chi = 0 \quad P_1\sigma_1 = P_2\sigma_2$,

(3) At $\chi = \ell \quad P_1\sigma_1 = \frac{-m\partial^2\xi_1}{\partial t^2}$.

Taking $B = B'e^{i\omega t}$, and applying the boundary conditions,

$$D_1 = D_3 ,$$

$$D_2 = iD_3 + \frac{\lambda B'\sigma_1}{\omega z} ,$$

$$D_3 = \lambda B'\sigma_1 \frac{(1 - \operatorname{Cos} K_1\ell + \frac{m\omega}{z} \operatorname{Sin} K_1\ell)}{(i\omega z - m\omega^2)(\operatorname{Cos} K_1\ell + i \operatorname{Sin} K_1\ell)} .$$

These relations determine the entire motion of the rod or tube. We are primarily interested in the motion of the concentrated mass, m, at $\chi = \ell$. Thus we evaluate

$$(A_1)_{\chi=\ell} = D_1 \operatorname{Cos} K_1\ell + D_2 \operatorname{Sin} K_1\ell$$

which, after some reduction, yields

$$(A_1)_{\chi=\ell} = - \frac{\lambda B'\sigma_1}{\omega z} \frac{(1 - \operatorname{Cos} K_1\ell + i \operatorname{Sin} K_1\ell)}{\left[1 + \left(\frac{m\omega}{z}\right)^2\right]^{1/2}} e^{i\theta}$$

where $\theta = \tan^{-1} \frac{z}{m\omega}$.

The acceleration of the mass, m, is then given by

$$(\alpha)_{\chi=\ell} = \left(\frac{\partial^2\xi}{\partial t^2}\right)_{\chi=\ell} = - \omega^2 (\xi)_{\chi=\ell}$$

$$= \frac{\lambda B'\sigma_1\omega}{z} \frac{(1 - \operatorname{Cos} K_1\ell + i \operatorname{Sin} K_1\ell)}{\left[1 + \left(\frac{m\omega}{z}\right)^2\right]^{1/2}} e^{i(\omega t+\theta)} .$$

Neglecting eddy currents, the total flux is related to the driving coil current by

$$B'\sigma_1 = \frac{LI}{N} ,$$

and if we neglect the resistance of the coil,

$$I = \frac{v}{i\omega L} ,$$

and we obtain

$$(\alpha)_{\chi=\ell} = \frac{\lambda v(1 - \operatorname{Cos} K_1\ell + i \operatorname{Sin} K_1\ell)}{Nz\left[1 + \left(\frac{m\omega}{z}\right)^2\right]^{1/2}} e^{i(\omega t+\theta-\pi/2)}$$

from which the constant voltage response of the transducer may be calculated.

This calculation has been made for a nickel tube of 0.650-in O.D., 0.020-in wall, 47 in long loaded with a mass of 25 grams, operating near magnetic saturation (B_o = 5100 gauss) with an alternating driving voltage of 30 volts rms. Total number of turns used was approximately 600. We have

$$\lambda = (-20)(10^3) \frac{\text{dynes}}{\text{gauss-cm}^2} , \quad (3)$$

v = 42(10^8) abvolts,

N = 600 tuns,

$$z = \rho_1 C_1 \sigma_1 = (8.7)(5)(10^5)(0.5) \doteq 13.05(10^5) \frac{\text{gm cm}^3}{\text{sec}},$$

m = 25 grams,

ℓ = 120 cm.

Since the phase relation between the acceleration and the impressed voltage is not of particular interest the calculation may be simplified to yield only the magnitude of α:

$$|\alpha| = \left[\frac{\lambda v}{Nz\left(1 + \frac{m^2\omega^2}{z^2}\right)^{1/2}}\right] \sqrt{2} \left(1 - \operatorname{Cos} \frac{\omega\ell}{C_1}\right)^{1/2} .$$

Inserting numerical values, and expressing in terms of frequency f,

$$|\alpha| = \frac{1.5 (10^5)}{(1 + 1.45 (10^{-8})f^2)^{1/2}} \{1 - \operatorname{Cos}[(1.5 (10^{-3}))f]\}^{1/2}.$$

This equation is plotted in Figure 3 (solid curve).

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DISCUSSION

Klingener, Sonotone Corp: I would like to know the audio power required to drive the magnetostriction vibrator?

Barnes: We use a 30-watt amplifier. In our preliminary work there were occasions when we blew fuzes on this amplifier, but we haven't done this lately.

Swartz, RCA, Camden: I notice on that frequency response figure that there seem to be very few frequencies showing up at the lower end of the spectrum. Is that a limitation of this jet system and if so how much of a low-frequency response can you get?

Barnes: The frequency there is a characteristic frequency of the jet, which is perhaps a somewhat fictional quantity. There are frequencies as low as four or five hundred cycles present in the actual vibration of the jet. We haven't measured much below this because we have used

a wave analyser, with the widest band width we could in order to try to get some average value for the spectral density. I am not sure that some of the assumptions we used in elucidating the data are much good below 500 cps. If you use narrower band widths, the meter starts jumping around and it's very difficult to obtain any really significant reading.

Orlacchio, Gulton Mfg. Corp: We have been playing around with vibrators for some time and have built a crystal vibrator that does have fairly good characteristics. One of the problems we always have with this type is to keep it compact so that we don't get any standing waves. Because of its length you probably do get a standing wave pattern in your vibrator, but the one we have been playing with is only about 2-in long. It is a crystal type piezoelectric vibrator which can be loaded up to two or three ounces on each side. It works out fairly well. We have been able to get a flat frequency response from 1,000 to about 9,000 cycles, within, I would say, plus or minus 10 percent. However, the problem here has been the matter of calibration. We have been using the interferometer technique and have found that it worked out very nicely. I would like to get your comments on how you calibrated your output.

Barnes: The measurements are within, I think, a reasonable degree of engineering accuracy. They are probably not good enough to satisfy a real scientist. The calibration was not actually accomplished. We used the accelerometers which were loaned to us by the Diamond Ordnance Fuze Laboratories. These were calibrated on their electrodynamic vibrator with which they have had a good deal of experience. They were calibrated at low frequencies with pretty good accuracy. Then their output was observed on a panoramic wave analyzer at frequencies up to 5,000 cps. They appeared pretty flat in this range so we extrapolated these data to 10,000 cps.

I would also like to mention here that the standing waves in this vibrator are really not a problem. The purpose of the termination is to absorb the energy which might produce such standing waves. There are a number of other things we could do to make even the theoretical output of this vibrator much flatter than it is.

Christensen, Firestone: Do you have any control of the frequencies you get with the jet, or is it just happenstance?

Barnes: I don't think we have enough data to indicate that we have much control.

Christensen: You spoke of the characteristic frequency of a jet. Do you have a wide variety of jets by type or serial number with which you can get certain patterns of frequencies?

Barnes: This characteristic frequency is a function of the diameter of the jet and the velocity of the jet. It can be varied over quite a range with a given jet.

Edelman, National Bureau of Standards: I wondered why you settled on the liquid jet. Have you also considered things like compressed air, superheated steam, or even gravel and sand as a means of getting a turbulent motion?

Barnes: I don't think a gas will actually do as well as a liquid. Gravel or sand or shot, something of that sort, could do quite well.

Edelman: You said you terminated your magneto rod with its characteristic impedance, in order to eliminate resonances as much as possible. Could you give us a rundown on how you get at the concept of the characteristic impedance of a rod like that, and how you pick a given material to terminate an end?

Barnes: The characteristic impedance I am talking about is the product of the density of the material, the velocity of sound, and the cross section area of the rod. This comes out, I think, pretty straightforwardly from the differential equation of motion. In order to get a simple solution it's very convenient to match the impedance at this junction because it eliminates reflections. In other words, the differential equation shows you that reflections are eliminated when this product is equal for the two materials of the junction.

DuBois, Boeing Aircraft: What materials did you consider for your termination?

Barnes: We actually used lead. As for other types of material, there may be some things which are better than lead. I think it's pretty obvious that lead has quite high elastic losses. We haven't carried out an investigation of other materials. Lead was very convenient to use.

* * *

BARIUM TITANATE VIBRATORS*

Seymour Edelman, Earle Jones, and Ernest R. Smith,
Diamond Ordnance Laboratories

A number of barium titanate vibrators have been built for simulating vibration environments. Any of these vibrators can subject complete components weighing up to 10 lb to accelerations of 10 g from 1,500 to 15,000 cps. Greater accelerations can be reached at the many axial resonances of each vibrator.

Adequate laboratory simulation of high-frequency vibration is handicapped by lack of suitable vibrators. It would be desirable to subject complete components to a considerable fraction of the amplitudes found in the field over the entire frequency range of interest and including bands of noise as well as pure frequency.

We are working toward this goal for loads of the order of 10 lb and the frequency range from 1,000 to 20,000 cps and we have reason to hope for success within a few years. However, our present state of progress is typified by the vibrators shown in the Figure. Barium titanate cylinders like those labelled 2 and 4 were made into the vibrators labelled 1 and 3, respectively, by attaching a close-fitting brass base and aluminum shake table to each with epoxy resin. Careful machining of the metal and careful lapping of the ceramic are necessary to ensure purely axial motion.

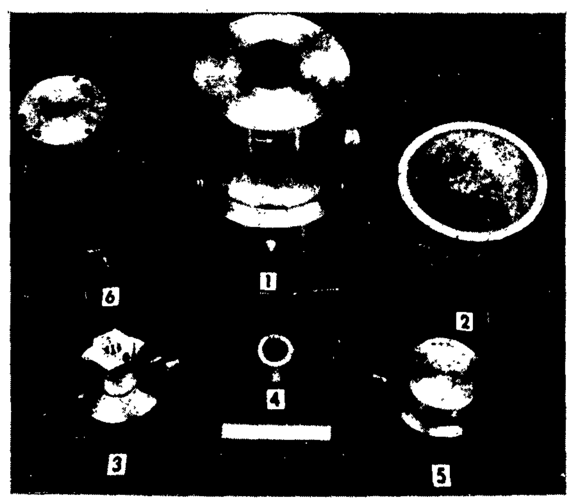


Figure 1 - NBS barium titanate vibrators

The vibrator labelled 6 is intermediate in size between 1 and 3. The vibrator labelled 5 illustrates the type of construction we think shows most hope for the future. The barium titanate element consists of disks cemented together with conducting epoxy resin. It is very stiff and not affected much by the size of the load but, since it is so short, the best output is in the ultrasonic range. To lower the range of effective operation the barium titanate portion must be longer; to provide low compliance to flexure it must be larger in diameter. So far, we have not succeeded in finding large diameter disks with sufficiently flat parallel sides and sufficiently homogeneous in behavior. A number of commercial suppliers, however, are working on the problem and we hope for a solution soon.

*This work was sponsored by the Diamond Ordnance Laboratories, Ordnance Corps, Department of the Army.

The type of vibrator labelled 1 is the most useful we have available at present. We have a number of similar cylinders from various suppliers. These differ slightly in dimensions, composition, and behavior. The one shown in the figure has resonances at 1,500, 4,250, 5,650, 7,300, 8,350, 10,000, 12,000, and 15,000 cps and antiresonances at 2,300, 5,300, 6,500, 7,700, 9,200, 11,000, and 14,000 cps when driving a concentrated 11 lb load.

Using the power amplifiers we have available and an inductance in parallel, it is possible to get at least 10 g at the antiresonances and several hundred g at some of the resonances. Since the resonances and antiresonances of the other shakers of the same type are at different frequencies it is possible to subject quite heavy loads to large values of acceleration at any selected frequency in the range.

By using feed-back from a vibration pickup whose response does not vary with frequency, mounted on the shake table to control the output of the amplifier, it should be possible to sweep through the frequency range at the constant value of acceleration represented by the lowest antiresonance, about 10 g for an 11-lb load. However, we have not done this since we hope that the vibrator of stacked disks will solve the problem more simply when we get it.

DISCUSSION

Christensen, Firestone, Missile Division: What are the pros and cons, the arguments between these disc vibrators and the hollow cylinders?

Edelman: The hollow cylinders are immediately available and the discs are not. The hollow cylinders give more motion. The discs are much stiffer and are less sensitive to the load. Even at resonances, applying a heavy load makes much less difference to a stack of flat discs than it does to a hollow tube. At present the only kind of a stack we have is the one that was shown in the figure. It has a very good high range, around 30 or 50 kc, but not much down in the audio range where we would like to have it.

Nitchie, ORL, Penn State: We have used a construction similar to the sandwich type for underwater acoustic transducers. My comment is relative to the lapping of the surfaces. We found that for our purposes we could get away with machining on the mating brass base an annular ridge a very few thousandths high, which ensured both mechanical and electrical contact with the face of the barium button. I seriously doubt if we are getting as good efficiency as you do with the mutually lapped faces. But at frequencies below transverse resonant frequencies of any of the components of the sandwich, we seem to be getting the whole signal.

Edelman: Well, we were driven to this lapping because when we first started gluing pick-ups together we found all kinds of funny bumps in the response curve. Also we found that we could get rid of these by just making the mating surfaces mate well enough. I can't say any more than that.

Barnes, Atlantic Research: I would like your opinion on just how necessary it is to use a conducting cement.

Edelman: Well, for many years we didn't and we got all kinds of horrified comments from various people because they claimed we were introducing a capacity voltage divider that got rid of a lot of our output. When the conducting epoxy became available we started using it and it did improve the output somewhat. I don't have the figures on just how much that improvement was, but my impression is that it was not very great. Perhaps Mr. Jones can say how much improvement was obtained by going from a nonconducting to a conducting epoxy?

Jones: The major advantage was an increase in the capacity. If you had a high-impedance voltmeter set up it didn't amount to very much. The open-circuit voltage remained about the same.

Hawkins, Sperry Gyroscope: What type of electrical equipment is needed to drive these vibration exciters?

Edelman: We have been using a MacIntosh 251 amplifier with a varied assortment of output transformers. Also we have been dickering with and trying to get some good output transformers made up for us. The problem is that at these high frequencies there is nothing available to give a lot of power and a good impedance match. So we have been using this 251 amplifier and trying various transformers in the output stage. Then we also use a coil, a conductance with a lot of taps on it at each frequency, to

adjust to electrical resonance between the coil and the capacitance of the shaker. In this way we get rid of a lot of harmonics and we also present a higher impedance to the power amplifier.

Hawkins: Are high voltages required?

Edelman: Yes, the voltage is the main thing with the barium titanate. That is, it takes voltages of the order of a thousand volts or so to get the kind of output I was talking about, several hundred g at the resonances.

Hawkins: I have another question. There is at least one manufacturer who makes something similar to this, Massa Laboratories. They use, I think, a different type of crystal.

Edelman: They use ADP length expanders, I think.

Hawkins: Do you know of any differences between the type of vibrating that you are describing at that type of thing?

Edelman: I believe that theirs are made up of a number of ADP length expanders side by side, and they are very good shakers for light loads. But they are not capable of vibrating heavy loads of the order of 10 lb or so unless Massa has changed things recently. I am not sure just what their present outlook is.

Siegelman, American Machine and Foundry Co: I was wondering what control you exercised over the dimensions of the outer radii to assure maximum deflection in the axial direction. I know that in the past it hasn't been possible to get from the manufacturers transducers of the exact dimensions we would like to get.

Edelman: As far as the shakers are concerned, we don't exercise any control at all. We take what we can get. The eccentricity doesn't seem to matter too much with the shakers. We are not bothered by axial resonances. As long as we have axial resonances we make use of them. There was one manufacturer who provided us with a very nicely rounded cylinder. But it was not as free of flexural motion as some we got from other manufacturers who were not as careful to give us a nicely rounded piece. We think that perhaps the reason is that when it is rounded to make it so nicely circular, in some places they took off the outer layer which was somewhat different in composition from the inner layers, and this affected its behavior.

George, General Electric Co: Regarding external influences such as temperature, humidity, etc., do you think you will have much trouble with this in the future? Can you do anything about it?

Edelman: Humidity is no great problem. Temperature is, if barium titanate is used. However, Mason at Bell Laboratories has worked out a number of barium titanate-calcium mixtures which have very low change of property with change of temperature. Mason has published this thing in "Acoustica," I think, and somewhere else, "Proceedings of the IRE," and you can find it there. And there are also newer compounds coming out, lead-zirconium-titanates, and similar things which are all directed towards this kind of problem. They are flat over wider temperatures and they work to higher temperatures than barium titanate.

Gorton, Pratt and Whitney: We used these ADP crystal shakers but for our purposes with a relatively high load trying to excite a small object at resonance, they are limited apparently by the mechanical strength of the crystal, either that or a flash over—you raise the voltage, it flashes over or somebody tells you to stop because the crystal is liable to break. What is this limit in the barium titanate as you raise the voltage?

Edelman: Well, we haven't reached the limit. Our limit has been the power amplifier. We haven't been able to do any damage mechanically or electrically. We have damaged a couple of ADP accelerometers by shaking them too hard, but the barium titanate is far stronger both electrically and mechanically than the ADP.

Schloss, David Taylor Model Basin: I had occasion to use the ADP exciters produced by Massa for testing of mountings, and as a matter of fact, I was using a static load of about 400 to 500 lb because the mechanical impedance, especially at the higher frequencies, is very low. But for the lower frequencies, so as not to run into trouble with noise, I have gone to 10,000 volts. And, as a matter of fact, I was only using about a 30-watt amplifier together with some very cheap transformers which I connected up, all the primary ones in a series and all the secondary ones in a series. In that way I could have got up to about 15,000 volts. But I believe the maximum voltage was only 10,000 volts. I couldn't get that voltage at the higher frequencies, but I didn't need it then, and to eliminate the trouble from some of the sources which I did get from the output at the lower frequencies, I used the narrow-band analysis.

Edelman: That is very interesting.

Stooksberry, Case Institute: I was wondering what set the minimum on your frequency range. There I realize you are interested in the higher frequencies, but why couldn't you extend it clear down to hundreds of cycles or even lower?

Edelman: Well, mostly because we are interested in acceleration. Displacement is obtained right down to DC, but at low frequencies the amount of acceleration that can be gotten from barium titanate drivers is very small.

* * *

A STEP FUNCTION ACCELERATION MACHINE FOR LIGHT-WEIGHT TEST SPECIMENS

George W. Schatz, Hughes Aircraft Co.

This paper describes an inexpensive shock machine developed for testing light weight items. The shock pulse obtained is approximately square-wave and a somewhat novel arresting media of lead and plastic is employed.

Present-day commercial shock machines are generally adequate for the loading functions and load capacities required for the majority of design test applications. Though the dynamic response of these machines, especially the medium- and high-impact variety, is quite random and erratic, the characteristic waveform (usually obtained with a high degree of filtering) is such that the pulse amplitude and total energy content conform to applicable design or specification requirements. Experience indicates that machines of this type have proved most useful in ensuring structural adequacy of airborne and transport equipment items.

There are many instances, however, where sufficient design information may not be obtained with commercial shock facilities. Such is certainly the case where the dynamic response of a test object to a specific and well defined forcing function is of primary importance rather than merely the test object's structural adequacy. A detailed example of such a condition is the test application for which the proposed machine was originally developed. Here the shock machine was to provide the following characteristics:

1. The machine was to be capable of subjecting the test specimen to a constant amplitude acceleration at any specified value between two and ten g. The amplitude could not vary more than ± ten percent from the specified nominal value.

2. The duration of the acceleration pulse at the specified amplitude was to be at least 30 milliseconds.

3. The rise time of the pulse (the time for the leading wave front to go from 0 to the specified amplitude) could not be greater than 3.0 milliseconds.

4. The foregoing requirements were to be met for a test specimen weight of up to three lb and with an over-all instrumentation flat frequency response from 0 to 500 cps.

In recent years, many attempts have been made to develop shock machines capable of meeting requirements such as these. Notable success has been achieved with machines which utilize hydraulic fluids to arrest the motion of a freely falling test specimen. However, low-impact hydraulic machines were not available commercially and their complexity and expense precluded the development and construction of a machine of this type for limited laboratory usage.

Basically all shock machines presently used may be divided into two categories:

- One, where the test specimen is secured to a mounting fixture or table and the fixture

or table is given a prescribed motion to produce the desired acceleration.

- Second, a type which uses a table or elevator which is set into motion by free fall or by a gradual application of force and then is brought rapidly to rest by allowing the elevator or table to strike a fluid, solid, or semisolid material.

Previous experience with machines of both types and of widely varying construction indicated two important characteristics. Machines of the first category were of such a structural or mechanical nature that their output appeared to be predominantly composed of random vibrations. The free-fall machines, on the other hand, were generally unsuitable for the production of a constant acceleration; however, they were for the most part much less complex and seemed to offer the best possibilities for minimizing waveform distortion due to structural vibrations. For this reason, a machine common to the latter category was selected as most practicable for the impulse characteristics specified above.

When used for small test specimens, the design and construction of the elevator and its supporting structure is relatively simple for a machine of this type and, as a consequence, complete attention could be given to the proper choice and utilization of the arresting media. Elastic arresting materials were immediately excluded since the resulting forces and hence the acceleration would be proportional to the displacement during impact rather than a constant. Likewise granular materials, such as sand, were excluded since, although far superior to elastic materials for generating constant acceleration, they nevertheless produce a retarding force which, probably due to impaction, is largely dependent upon the displacement. The use of friction devices, such as brake bands, was excluded for the same reasons as were hydraulic devices, namely, design complexity and expense.

Obviously, the most direct solution to a suitable material for a step or constant acceleration drop machine would be a material which deforms plastically under impact. The most significant objection to such a material is the probability of viscous or strain-rate effects influencing the waveform during the initial phase of contact. Due to the fact that some degree of success in producing short-duration square wave accelerations had been attained with commercial machines which used a steel punch to penetrate a block of lead, lead was decided upon as the first material to be tried. Since a long-duration, as well as a constant acceleration was required, a simply supported lead beam was used rather than a block of lead and a penetrating device. The beam was so supported that a deflection of approximately two inches at the center span was possible, thus allowing unusually long durations at low-acceleration amplitudes.

The result of early experiments showed that after initial contact the amplitude of the shock pulse was, for all practical purposes, constant. However, the leading wave front was extremely steep and was composed of high-amplitude, high-frequency vibration lasting about three milliseconds. It is believed that this vibration was excited primarily by the coincidence of the steep wave front with the dominant structural modes of the machine.

Further experimentation resulted in the use of a dough-consistency plastic which, when placed between the bottom of the elevator and the beam, acted to control the rise time of the pulse in such a way that the resonant modes of the machine were not significantly disturbed. A convenient plastic was found to be available in the form of cylindrical pellets 1-in long and 1-1/2-in in diameter. These were obtained in this form from the Presstite Engineering Company, St. Louis, Mo. The beams used were 4-in wide and 1/4-in thick. Using the above pellets and lead beam, the results illustrated in Figures 1 through 4 were obtained. Though the

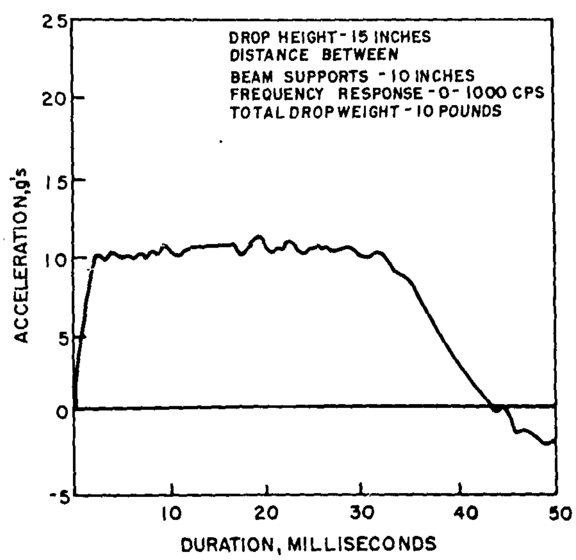


Figure 1

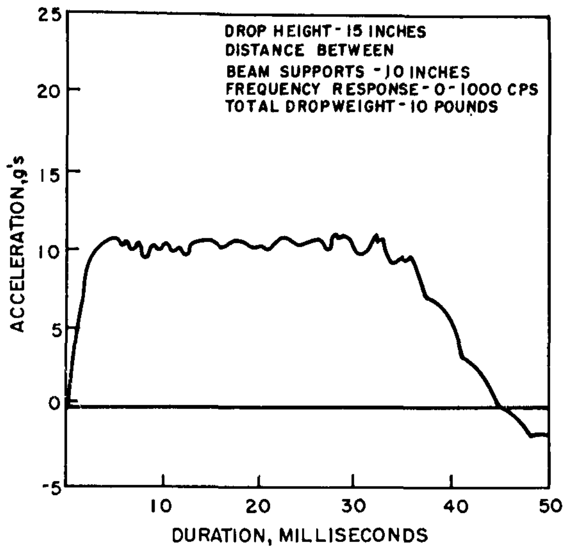


Figure 2

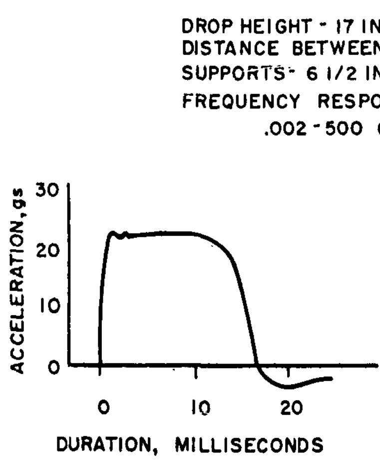


Figure 3

excellent rise-time characteristics provided by the beam alone were necessarily sacrificed somewhat, these figures show that both the amplitude and rise-time characteristics of the pulse are considerably better than the first three requirements.

It is also interesting to note that the form of the trailing wave, especially at the higher acceleration levels, is such that the total impulse approximates a square wave quite well. Figures 1 and 2 were obtained with Consolidated Engineering Recording oscillograph equipment and Figures 3 and 4 with a Polaroid Land Camera and Dumont Oscilloscope. In all cases, the

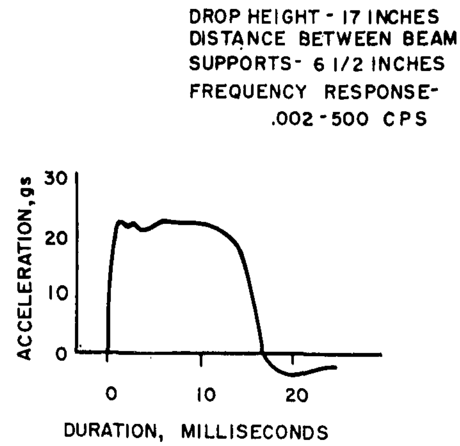


Figure 4

transducer and the excitation for the transducer were the same, namely a model ASA-100-200 Statham strain gage accelerometer and a Consolidated Engineering carrier amplifier system.

Figure 5 illustrates the general configuration of the machine and shows the relative positions of the elevator, the plastic, and the lead beam prior to impact. The elevator and the vertical standoff, which is welded directly to the top center of the elevator, are constructed of aluminum. In initial applications, the test specimen was secured to one face of the standoff and the acceleration transducer to the other face. A tee-shaped tup with a cylindrical striking edge is screwed into the bottom center of the elevator and is the only portion of the elevator assembly which contacts the plastic. The tup is so oriented that its striking edge is parallel to the surface of the beam and perpendicular to the beam's longitudinal center line. With this arrangement, tups having different striking edge configurations may be used to alter the shape of the leading wave front, if so desired, without resorting to changes in any of the other drop parameters.

A structure for supporting the lead beam is bolted directly to the base of the drop machine. This device consists of two knife edges which are carried in milled slots and which are separated and adjusted by means of a threaded rod. When the distance between the knife edges has been set to the required value, the knife edges may be locked into position with set screws. Details of the tup and beam supporting structure are illustrated in Figure 6.

A plate with a microswitch attached is also shown in the figures. This switch is used to

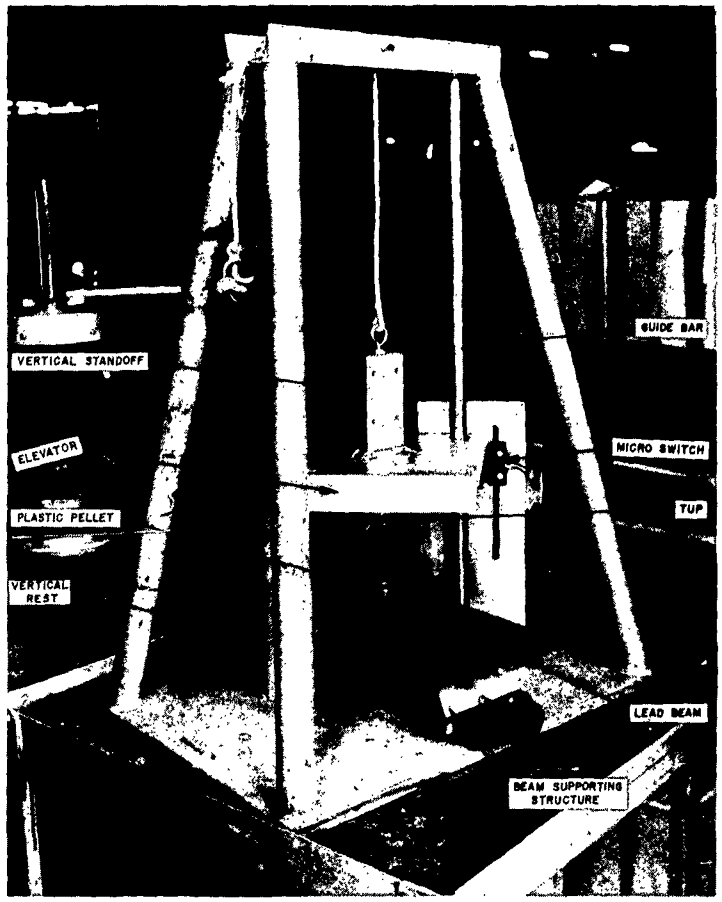


Figure 5 - General configuration of step-function acceleration machine for testing light-weight material

trigger a dual beam oscilloscope (when used) just prior to impact. In this way, a permanent record of the acceleration input to the test object as well as the test object's electrical response to the impulse may be obtained.

Proper orientation of the elevator assembly relative to the beam is maintained by two vertical guide bars over which slide two bronze bushings pressed into opposite corners of the elevator. The elevator may be maintained at a sufficient height to allow adjustments of the pellet and beam to be made by using a vertical rest positioned between the elevator and the base of the machine as shown in Figure 5. If the combined weight of the elevator and test specimen is not excessive, a rest equal in length to the desired drop height may be used. In such a case, the elevator may be held for a brief moment after the rest is removed and then released by hand.

The entire machine is approximately 28-in high and occupies a floor or bench space of approximately 14 x 20 in. The maximum drop height is 18 in.

The present machine was developed for very specific and limited test purposes. As a consequence, certain inconveniences associated with the operation of the machine were not given a great deal of attention. For example, each time a drop is completed, the lead beam must be removed and straightened, either by hand or in a suitable press. The encumbrance of a lead beam of sufficient capacity to produce the same amplitude and duration characteristics on large specimens as have been produced for the light-weight specimens (5 lb or less) tested thus far would be intolerable.

It will also be noticed from the differences between Figures 1 and 2, and Figures 3 and 4,

distortion is difficult and yet vital to a proper leading wave shape and reproducibility.

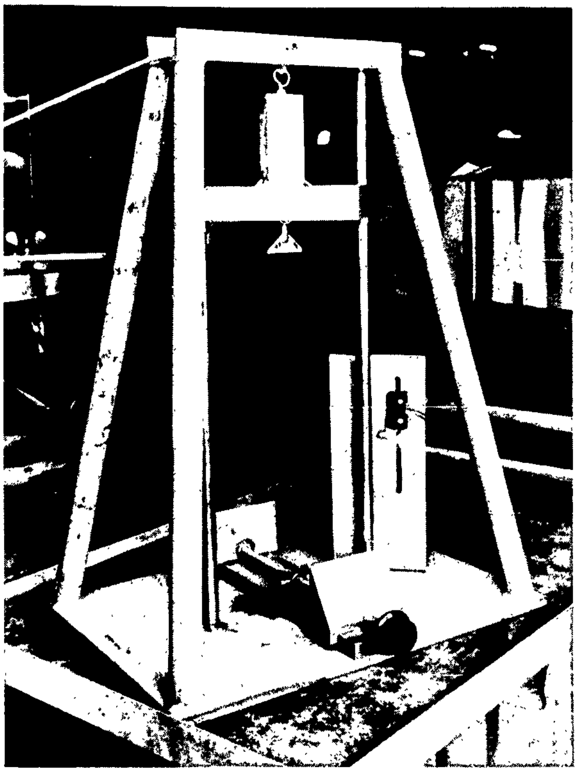


Figure 6 - Tup and beam-supporting structure of step-function acceleration machine

The results of environmental testing on which the machine has been employed for over two years indicate many advantages and features not obtainable with present commercial machines. The more significant features are:

1. The shock is produced with simple and inexpensive apparatus. The construction details illustrated in Figures 5 and 6 should give a good indication of the machine's inexpense relative to existing precision acceleration machines.
2. The shock pulse is simple in nature and capable of being specified by a simple mathematical curve. Such waveforms permit ease of interpretation of results and mathematical analysis of the test specimen's response.
3. The shock is uniformly reproducible, i.e., repetitive, and reasonably independent of the test specimen. Uniform reproducibility is especially significant in that results obtained on different test objects and in different test laboratories may be easily compared.

The results of tests performed in an effort to extend the capabilities of the machine beyond the two to ten g range specified above have shown that amplitudes in excess of 20 g with durations of 20 milliseconds and amplitudes less than 2 g with durations in excess of 100 milliseconds or any consistent combination of amplitude and duration within these limits can be produced with the present apparatus.

that the waveform is not entirely independent of the test object. However, the test object related to Figures 3 and 4 was, for the most part, inert while the specimen related to Figures 1 and 2 consisted of a number of spring mass systems. Also, the wider band pass used in Figures 1 and 2 is undoubtedly a contributing factor to these differences.

Attempts to deviate significantly from the amplitude and duration values expressed previously will generally result in tedious calibration programs and the use of beam, pellet, and tup configurations substantially different from those presented here. In addition, the protection of plastic material from age hardening and distortion is difficult and yet vital to a proper leading wave shape and reproducibility.

In conclusion, though significant disadvantages exist, it is felt that the principal objective in discussing a machine of this nature is not to present an inexpensive and compact square-wave impulse device of universal application but rather to show what can be obtained with arresting media which deform inelastically and which possess the damping or energy dissipation capacity comparable to that exhibited by the plastic and lead utilized in the present machine.

* * *

DETERMINATION OF TERRIER SHOCK AND VIBRATION ENVIRONMENT

A. Fine, T. Whiteley, D. Bell, and M. Buus, NOL, Corona

A TERRIER Missile was fired in a captive test stand and later in free flight. The shock and vibration information obtained was reduced by analogue and digital methods. A description of the test stand is given and comparisons of the shock and vibration information are made. The problems are summarized together with suggestions for the future program.

INTRODUCTION

In order to determine the environment induced in the TERRIER Missile by sustainer burning, it was decided that the restrained firing or captive test would be used. In a captive test program, the missile is not expended, and it can be used several times by simply supplying new sustainer motor units. The shock, vibration, and temperature data can be obtained during the sustainer burning with all the components of the missile system functioning. The same missile, termed the Vibration Analysis Vehicle (VAV), was subsequently expended in a free-flight test. The telemetered environmental data obtained from the VAV could be compared with the data from the captive test, and thereby the degree of correlation of the data could be determined.

The Captive test and VAV programs were established by the Bureau of Ordnance as a joint effort under the technical direction of Convair, Pomona, with participation by U. S. Naval Ordnance Laboratory, Corona, Calif., and U. S. Naval Ordnance Test Station, China Lake, Calif. U. S. Naval Air Missile Test Center, Pt. Mugu, Calif., contributed a great deal of help, information, and experience to the Captive test program of the TERRIER missile.

AIR-SPRING AND CAPTIVE STAND

In the Captive test program, the thrust was neutralized by means of an air-spring or air-buffer. This air-spring was a modified oleo-strut. The modifications, as shown in Figure 1, were made by U. S. NAMTC, Pt. Mugu.

Since there was some concern whether or not the TERRIER Missile skin would tolerate the forces exerted upon it, a special adapter was made to strengthen the nose of the missile, as shown in Figure 1. This adapter was fastened to the missile by means of set screws. A hole was provided to permit the entry of an electrical cable to the forward section of the missile for instrumentation. Tests were performed to determine if a load acting perpendicularly to the piston arm would cause binding of the piston, however, no tendency to bind was indicated.

The test stand was made as two separate structures; one to support the missile, and the other to withstand the thrust of the sustainer motor. Each was constructed of 12 x 12-in timbers anchored in concrete. The missile was fastened to the support stand by 8 nylon ropes which were attached as follows: 4 to the forward section at Station 45; 4 to the rear

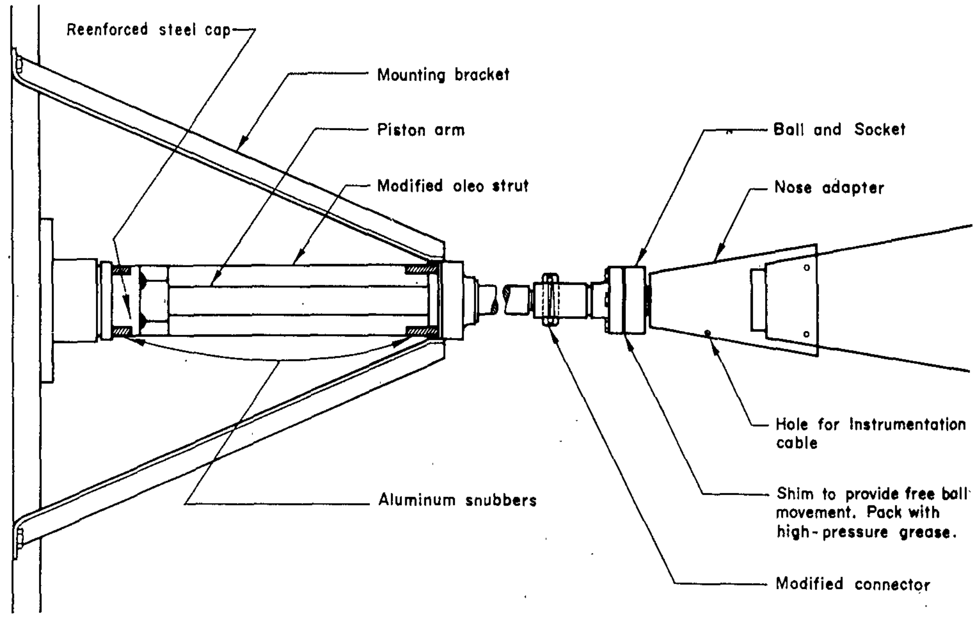


Figure 1 - Thrust neutralizer assembly captive missile shock stand

section at Station 152. The stations are designated in inches from a reference point (Figure 2). Each nylon rope was fastened by means of a clevis arrangement to a steel girdle that securely encompassed the missile. A German splice was used in securing the nylon ropes and turn buckles were provided for tightening. A German splice is a special splice that will not slip with nylon. Figure 3 illustrates the attachment to the missile, and Figure 4 shows the forward part of the TERRIER Missile with the thrust stand arrangement. A water pipe was attached to the stand in such a manner that it could be pivoted manually into position after firing of the sustainer

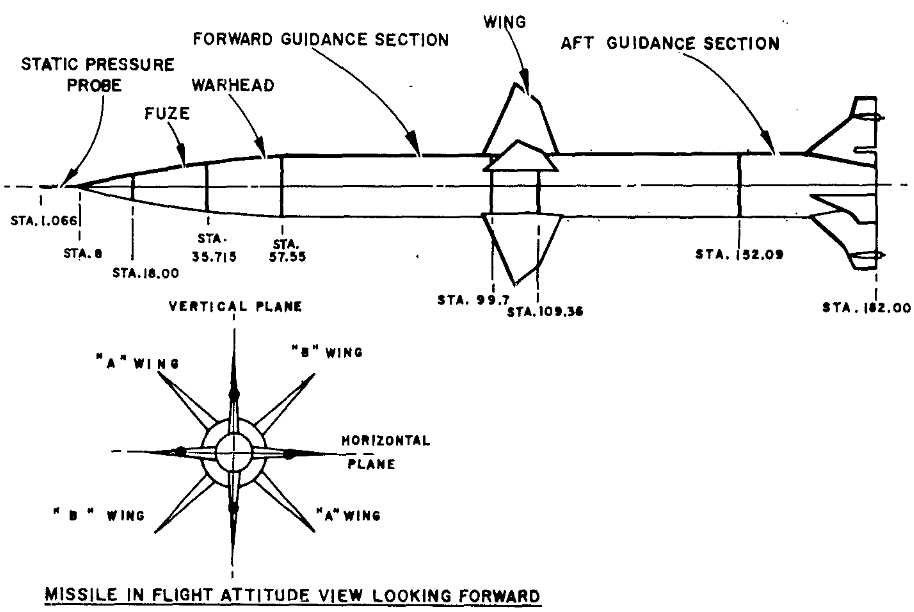


Figure 2 - Views of Terrier Missile

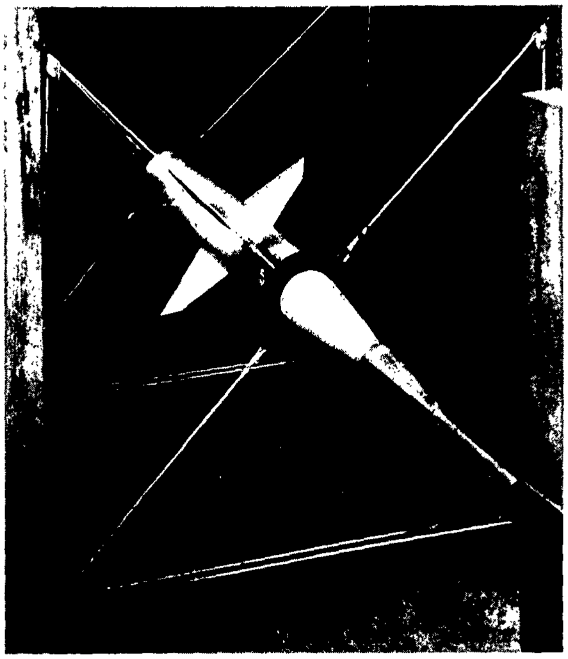


Figure 3 - Missile suspension system

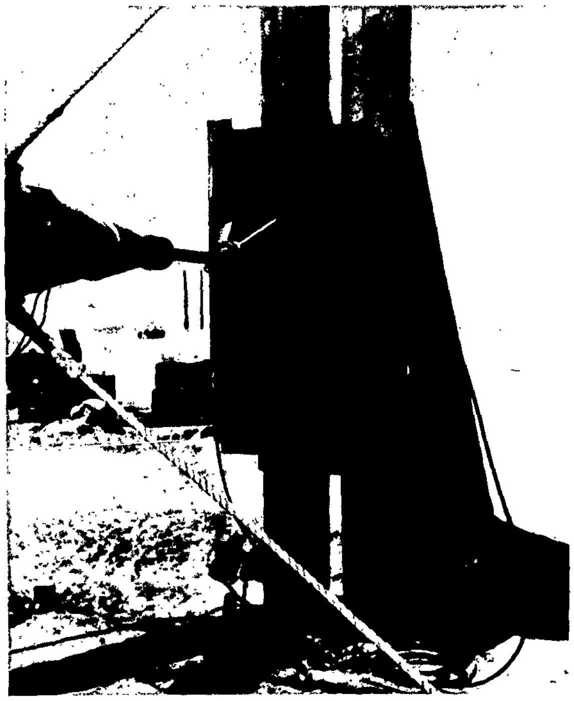


Figure 4 - Thrust stand

motor, and eject water into the blast tube to lower the temperature, thus preventing transfer of heat that would damage adjacent components.

INSTRUMENTATION

The standard missile configuration was modified to include twelve barium titanate accelerometers as well as the associated amplifiers and electrical wiring located at various stations throughout the missile. The exercise warhead contained 3 Rosen Model 960 crystal-controlled, frequency-modulated transmitters in a specially built package. The 3 RF outputs were coupled by a Bendix Quadruplex Unit TNC-10 to the waveguide and antenna on missile fin No. 4. The 9 Rosen Model 957 subcarrier oscillators, associated with the warhead telepack, were used for the 3 forward instrumentation locations. At each of the 3 general station locations, 3 accelerometers were mounted with axes in mutually perpendicular planes. Convenient structural members were used in mounting the accelerometers. The standard Rosen telepack, in the missile boattail, was used for the 3 accelerometers located in the aft section.

In each of the 4 telepack transmitters, the 70-kc subcarrier oscillator channels were used for the longitudinal accelerometers, the 40-kc subcarrier channels were used for "B" wing accelerometers, and the 22-kc subcarrier channels were used for the "A" wing accelerometers. "Longitudinal" means that the sensitive axis of the accelerometer was parallel to the longitudinal axis of the missile. "A" wing and "B" wing mean that the sensitive axes of the accelerometers were parallel to the corresponding axes of the wing rotation.

Figure 5 shows the orientation and station location of the instrumentation accelerometers. In one of the tests, the 3 accelerometers in the forward section were Endevco Model 2203 and the remaining 9 accelerometers were Glenite Model A-320T, and in another test they were all Endevco accelerometers. The Endevco Model 2607 cathode-follower amplifiers were used with all accelerometers. Included in the system were low-pass filters preceding the amplifiers and bandpass filters following the amplifiers.

For the Captive ground tests, vibration channel landlines paralleled the telemetry channels. The landlines were electrically matched to an instrumentation van for recording. Recordings were made simultaneously in the van on Consolidated Engineering oscillographs and Ampex magnetic tape recorders.

For the Captive ground test and the flight test the NOTS Telemeter Ground Stations were used for the RF link. Test information was recorded on Miller recorders and Ampex magnetic tape recorders.

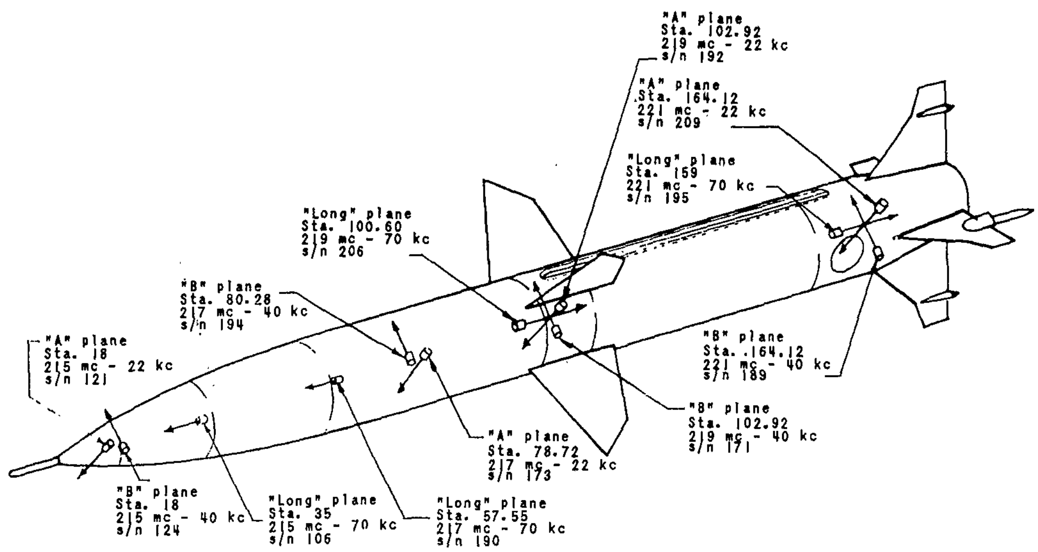


Figure 5 - Missile C-256 flight vibration instrumentation

THE ANALOGUE METHOD OF DATA REDUCTION

The initial reduction of the magnetic tapes containing vibration information from the missile tests was done by the Measurements Branch of the Component Test Department at the U. S. NAMTC, Point Mugu, Calif. The first step in the reduction of all magnetic tapes was pre-editing. In pre-editing, a tape was played back on an Ampex 500 with the signal displayed on an oscilloscope. A number of pre-editing runs were made to determine the frequency response, amplitude, distortion, signal-to-noise ratio, and resolution. All of these factors were known for each channel of information received on one RF channel before the proper adjustments were made on the re-recording equipment.

This information was reproduced on a record other than magnetic tape to permit visual observation of the entire test. An example of this reproduction is the Miller record. Figure 6 is a block diagram showing the major components of the system used to produce Miller oscillograph recordings.

Figure 7 illustrates the composite signal plus high and low gain of the analyzed frequency. This permitted direct visual comparison of various station outputs. If a known event such as the chugging of the Autopack or sustainer ignition can be located on the composite wave, the frequency at which maximum deflection occurs can be easily located. Autopack chugging is a periodic occurrence resulting from the pneumatic-hydraulic system.

Each Miller record has on it the flat-frequency response calibrations for the particular channels on that record. These calibrations were produced using the following method. The ac field calibrations were played back on the Ampex 500 and the output plotted in the form of frequency versus output in decibels, known as a frequency-response curve, for the entire instrumentation system, see Figure 8.

The zero point of frequency which was used as a basis for all other frequencies usually occurred at 200 cps. This particular frequency, 200 cps, is the point at which maximum output occurred for the instrumentation system installed in the missile. The amount of attenuation in decibels required to bring the response level to the base level can be selected from the curve for each particular bandwidth center frequency being analyzed.

The calibrations that were recorded at the beginning of the analyzed frequency record, were made using the same base. By this method, the analyzed frequency records were produced having a flat frequency response through the entire range analyzed. Step-frequency analyses (Miller records) were made from the magnetic tapes using a 20 cps bandpass filter up to 100 cps,

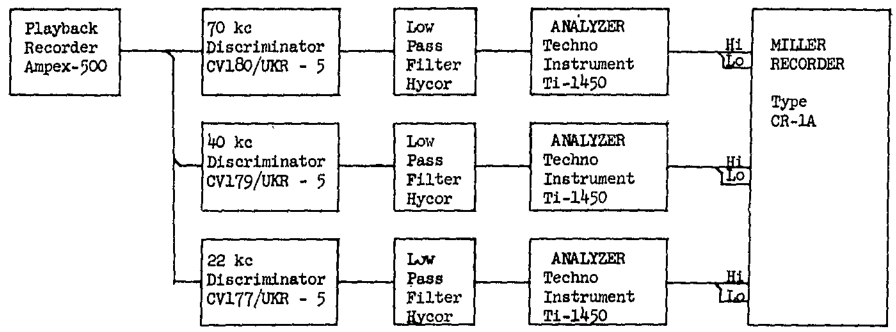


Figure 6 - Block diagram of magnetic tape reproduction on Miller record. (The analyzers are corrected to 1/10 of 1 db for flat frequency response. The value of the low pass filters used, when making the analyzed frequency recording, is immaterial except that it must be higher than the frequency being analyzed. Standard values for the low pass filters are used when making the composite record: Low pass filter for composite only: -70 kc - 6,300 cps; 40 kc - 3,600 cps; 22 kc - 1,980 cps. These values may vary slightly depending on the condition of the information on the magnetic tape.)

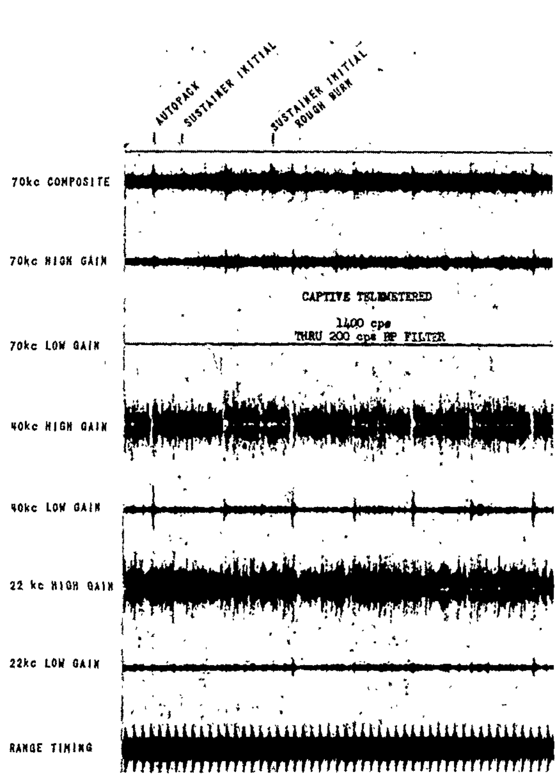


Figure 7 - Miller oscillograph record

and a 200 cps bandpass filter from 200 cps through 4,000 cps.

The Miller recordings were analyzed by the U. S. Naval Ordnance Laboratory, Corona, Calif.

The events of interest were selected from the composite wave. The individual events chosen for study occurred at approximately the same time on each RF channel for the same flight or test. A variable time difference was noted between station locations of the accelerometers because of shock transmission time through different parts of the missile. Each roll of analyzed frequencies contained information from the three mutually perpendicular accelerometers located in the same area at approximately the same station. There were two waveforms for each accelerometer on the Miller records, one high gain and one low gain. Because of the limited space of the Miller record, the high gain became saturated at about the 5 g level. The vibration forces in excess of the 5 g level were obtained from the low-gain wave. Each waveform was allowed 15/16 in before saturation occurred. This gave 6 waveforms of analyzed frequency plus the composite wave of one accelerometer, usually the longitudinal, and the range timing on the Miller record for each bandwidth center frequency that was analyzed as shown in Figure 7.

The calibrations at the beginning of the record were scaled manually and recorded with the millivolts required to produce each deflection. From these calibrations, a curve was drawn for each subcarrier channel using the amplitude of the wave as the ordinate and millivolts as the abcissa. The millivolt values were converted to g units by means of the accelerometer calibrations supplied by the contractor, see Figure 9.

At the chosen events, the peak-to-peak deflections of each of the waveforms were scaled

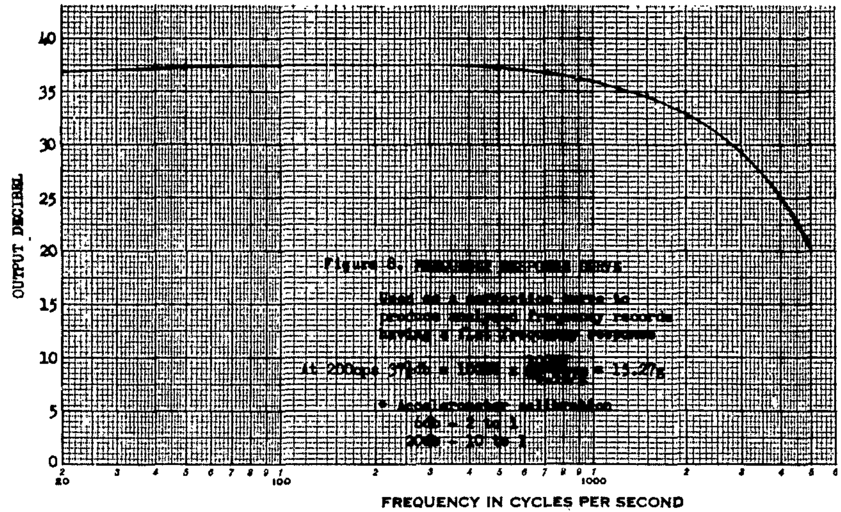


Figure 8

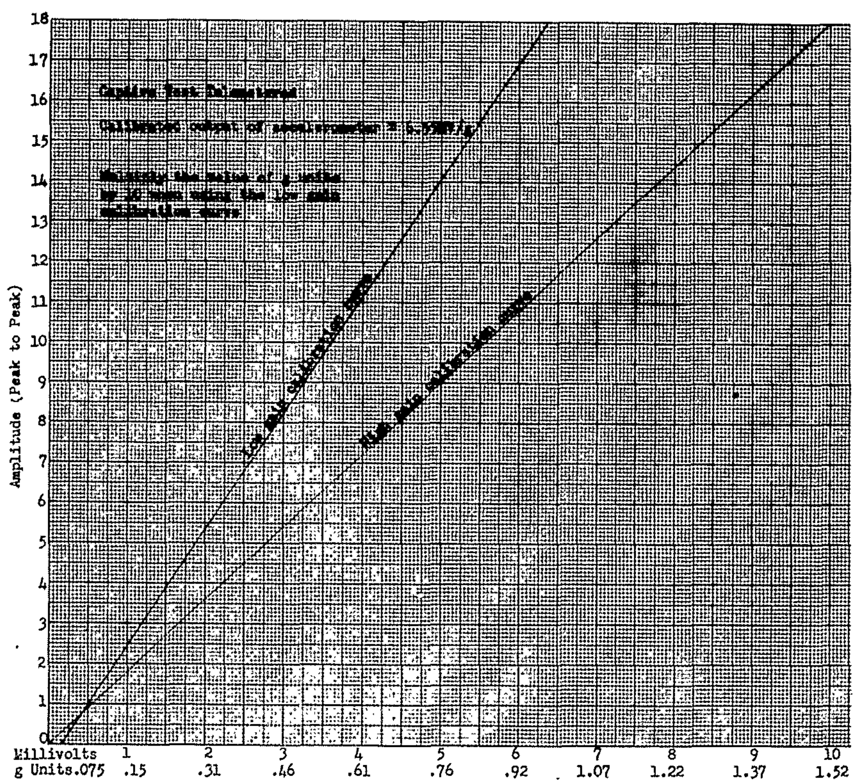


Figure 9

and the readings tabulated. The amplitudes, as tabulated, were found on the calibration curve and the value of g was obtained. The value of g was entered on a second set of tabulated sheets, see Table 1.

For a missile undergoing both captive and free-flight tests, there were three complete sets of records. The captive landline and captive telemetered information was compared. The telemetered information from the flight was compared with the captive telemetered information to find the correlation between captive tests and free flight. Figure 10 is a portion of the 1,000 cps analyzed frequency cut of the Miller record showing the Autopack chugging, sustainer ignition, and the initial rough burn of the sustainer motor. The Autopack chugging which was read for the flight was not shown because it was read prior to booster ignition. However, the Autopack chugging was noticeable throughout the entire flight. All information shown in Figure 11 was received from the same accelerometer which was located aft of the warhead and forward of the sustainer in a plane perpendicular to the longitudinal axis.

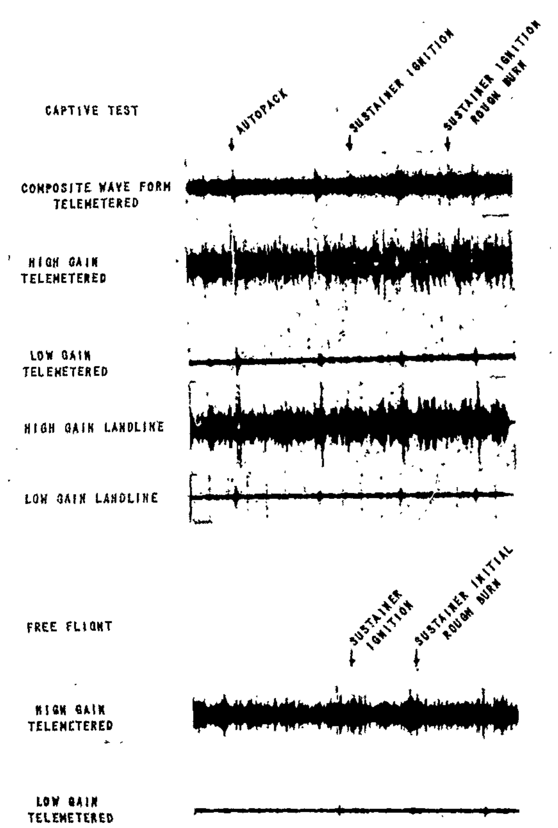


Figure 10

The g values as recorded in Figure 10 were plotted with frequency as the abcissa. Three curves were drawn on each graph which constitutes one event for one accelerometer for one missile. The three curves were plotted from information received from captive landline, captive telemetered, and flight telemetered. This reduced the 8,712 tabulated readings to 72 sets of curves which were easier to examine and analyze.

Figure 11, 12 and 13 are the final curves used to display the maximum g value at various frequencies. Because the data were obtained from readings taken at the center frequency, the true graph from the data would be in step form. However, the continuous-line curve is better suited for comparative analyses and therefore was used. The maximum values remain true within the error limits of the calculations and instrumentation. It would be desirable to display a three-dimensional approach which would cover the force, frequency, and duration. However, there was no method available at the time the records were reduced and analyzed to accomplish this. Therefore, no curve could be plotted showing the duration of a vibration at a specific frequency.

THE DIGITAL METHOD OF DATA REDUCTION

A given vibration environment may be identified by specifying the vibration frequencies and the amplitude distribution, i.e., the relative amount of time at various levels of acceleration. The distribution of amplitude levels is of importance because many types of equipment will withstand short-duration, high-level shocks, but will fail if these shocks are repeated a sufficient number of times. A quantitative knowledge of the distribution should be of great value in determining whether or not a vibration environment will produce adverse effects.

The electrical signal containing the vibration information is analyzed, or decomposed, into various frequency bands in a manner similar to the method previously described. A particular event or portion of flight may be selected for study by time-gating the signal or by cutting out the desired portion of magnetic tape and forming a continuous loop. The system digitalizes the filter output signal and electronically counts the number of peaks in selected amplitude ranges. This provides not only a measure of maximum peak amplitude, but accurate digital data for obtaining a statistical distribution of the acceleration levels existing in a given vibration.

TABLE 1
Vibration Data*

Comparative Tabulations of: Flight Telemetered, Captive Telemetered, Captive Landline

Event	Range Time	Type	20 cps Band Pass Filter					200 cps Band Pass Filter																	
			20	40	60	80	100	200	400	600	800	1,000	1,200	1,400	1,600	1,800	2,000	2,200	2,400	2,600	2,800	3,000	3,200	3,400	3,600
Autopack	-0.84	Flight	0.14	0.14	0.14	0.22	0.66	1.3	1.59	1.19	0.32	0.53	5.1	6.2	2.0	4.0	4.0	9.5	14.1	5.1	5.1	6.2	2.54	1.9	4.0
	0.9	Cap. T/M	0.18	0.18	0.18	0.25	0.45	3.3	2.0	2.9	1.02	2.0	4.4	7.6	2.9	2.9	5.6	11.7	6.9	2.9	3.7	3.7	2.0	5.3	6.0
	0.9	Cap. L.L.	0.10	0.13	0.13	0.27	0.27	1.62	1.44	1.13	1.39	1.47	1.8	3.6	2.1	1.58	3.3	6.6	4.2	2.1	1.6	2.7	2.1	2.1	4.4
Sustainer Ignition	4.61	Flight	0.60	0.90	0.30	0.45	0.74	1.04	0.74	0.30	0.45	1.49	2.54	2.69	1.34	2.54	6.2	14.1	4.0	2.84	2.69	2.24	1.64	2.69	2.24
	1.325	Cap. T/M	0.18	0.22	0.10	0.18	0.14	0.18	0.25	0.46	0.18	0.80	1.22	1.22	0.53	2.9	6.0	12.8	6.0	2.0	1.09	4.4	3.3	5.3	6.0
	1.325	Cap. L.L.	0.23	0.23	0.32	0.23	0.41	1.58	1.52	1.58	0.76	1.61	3.3	3.9	1.6	1.58	4.4	7.6	5.5	2.1	2.1	3.6	1.6	1.58	3.0
Sustainer Initial Roughburn	4.88	Flight	0.60	0.45	0.44	0.29	0.75	0.44	0.44	0.29	0.44	0.89	2.24	2.69	1.49	1.79	8.0	11.5	4.0	2.84	2.24	1.49	2.09	1.49	1.79
	2.425	Cap. T/M	0.10	0.10	0.10	0.10	0.10	0.18	0.18	0.18	0.25	0.88	2.0	2.0	0.73	2.0	5.3	9.8	6.0	2.9	1.25	0.66	0.88	1.02	0.80
	2.425	Cap. L.L.	0.10	0.13	0.13	0.10	0.45	0.76	0.37	0.41	0.45	1.0	1.58	1.60	1.52	1.61	4.2	5.7	4.9	2.7	1.6	1.52	1.39	1.35	2.7
Sustainer Mid-Burn	11.30	Flight	0.45	0.45	0.30	0.30	0.75	2.24	1.49	0.30	2.54	1.49	6.2	4.0	0.90	2.69	6.2	14.1	9.2	3.0	2.84	1.49	1.79	2.24	1.94
	14.005	Cap. T/M	0.18	0.14	0.10	0.10	0.80	0.32	0.18	2.0	1.22	0.80	4.4	4.4	2.9	2.9	6.9	8.6	6.0	4.4	3.7	2.9	2.9	6.5	5.3
	14.005	Cap. L.L.	0.18	0.13	0.18	0.18	0.58	1.6	0.86	1.0	0.37	0.68	1.8	3.3	2.7	1.58	3.9	5.7	4.2	2.7	2.7	1.6	1.21	2.7	2.7
Sustainer Major Shock	9.40	Flight	0.45	0.45	0.30	0.60	1.19	2.69	1.79	0.45	2.54	2.24	2.54	7.0	1.04	2.54	4.0	12.6	9.2	2.84	1.94	1.49	1.79	4.0	1.94
	7.825	Cap. T/M	0.10	0.10	0.10	0.32	0.80	3.7	2.9	2.0	0.66	0.66	1.02	1.02	2.9	2.9	5.3	12.8	5.3	2.45	3.7	0.88	2.9	6.0	3.7
	7.825	Cap. L.L.	0.32	0.27	0.18	0.27	0.63	1.52	1.52	1.31	1.58	1.52	3.3	3.0	2.1	1.58	3.6	6.6	4.2	2.4	3.3	1.6	1.6	3.6	1.58
Sustainer Burnout	+27.13	Flight	0.36	0.30	0.30	0.45	1.04	0.30	0.45	0.30	0.45	0.90	1.79	2.24	0.90	2.84	12.6	10.4	9.2	3.0	2.09	1.34	1.79	1.94	2.54
	23.805	Cap. T/M	0.32	0.25	0.14	0.18	0.39	2.9	2.9	1.15	1.02	0.80	3.7	6.9	2.0	3.7	10.7	11.7	7.6	2.45	2.0	0.67	0.73	1.02	0.80
	23.805	Cap. L.L.	0.50	0.41	0.13	0.27	0.32	0.63	1.39	1.61	0.72	1.03	3.3	1.58	2.1	1.6	4.9	7.6	4.4	1.6	1.58	1.6	1.8	1.6	1.08

*MISSILE C-256, Station 80.28; B wing, 217 mc, 40 kc
All values expressed in G units
Any value of g below 0.5 may be in error 0.2 g

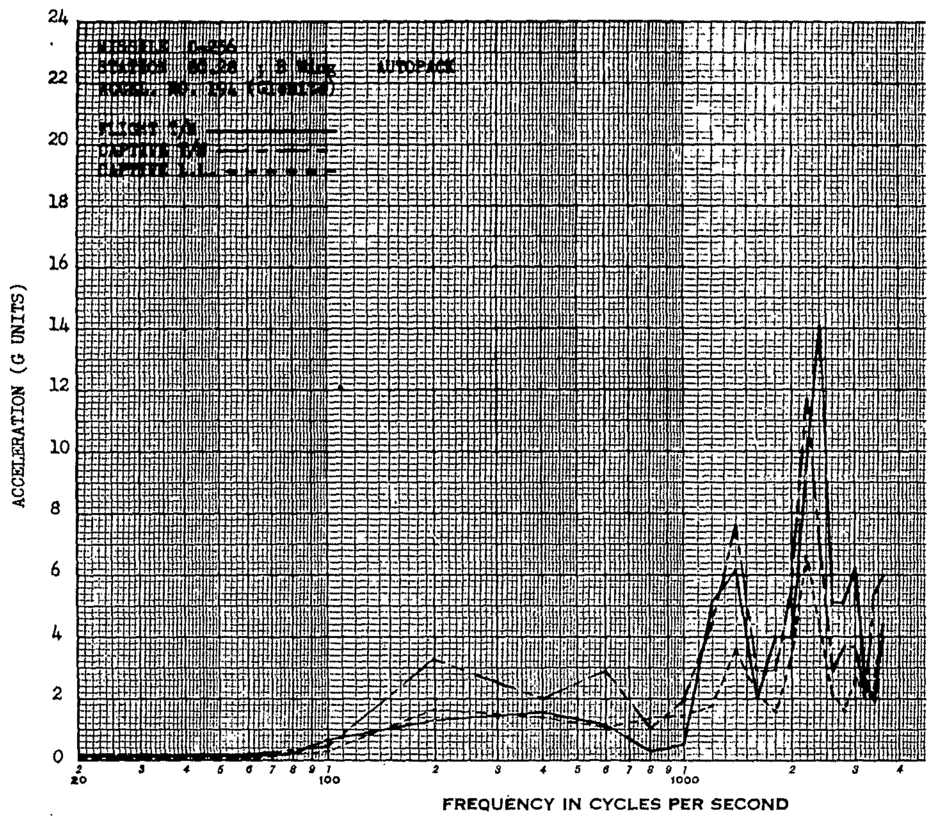


Figure 11

In brief, the system consists of the following components.

1. A variable-frequency electronic filter for selecting the frequency band to be analyzed.
2. An instrument specially designed to sample the vibration waveform at each peak, producing regularly shaped pulses equal in amplitude to the peaks of the input signal. This instrument is described later in the paper and is the only equipment required for the system that is not commercially available.
3. A pulse-height analyzer of the type used in nuclear work for "sorting" pulses into groups having amplitudes which fall within selected limits.
4. An electronic counter which counts the number of pulses in the amplitude group passed by the pulse-height analyzer.

The system is readily capable of analyzing frequencies up to 10 kc per sec. The range from 30 to 10,000 cps was divided arbitrarily into 14 bands, as shown in Table 2, with band limits in the ratio approximately 1.5 to 1. The filter used provides 24 db per octave cutoff at each side of the band.

TABLE 2
List of Filter Bandwidths Used

Center frequency (cps)	Lower limit (cps)	Upper limit (cps)
39	30	50
61	50	75
95	75	120
147	120	180
221	180	270
330	270	400
490	400	600
735	600	900
1,110	900	1,350
1,650	1,350	2,000
2,460	2,000	3,000
3,680	3,000	4,500
5,550	4,500	6,750
8,280	6,750	10,000

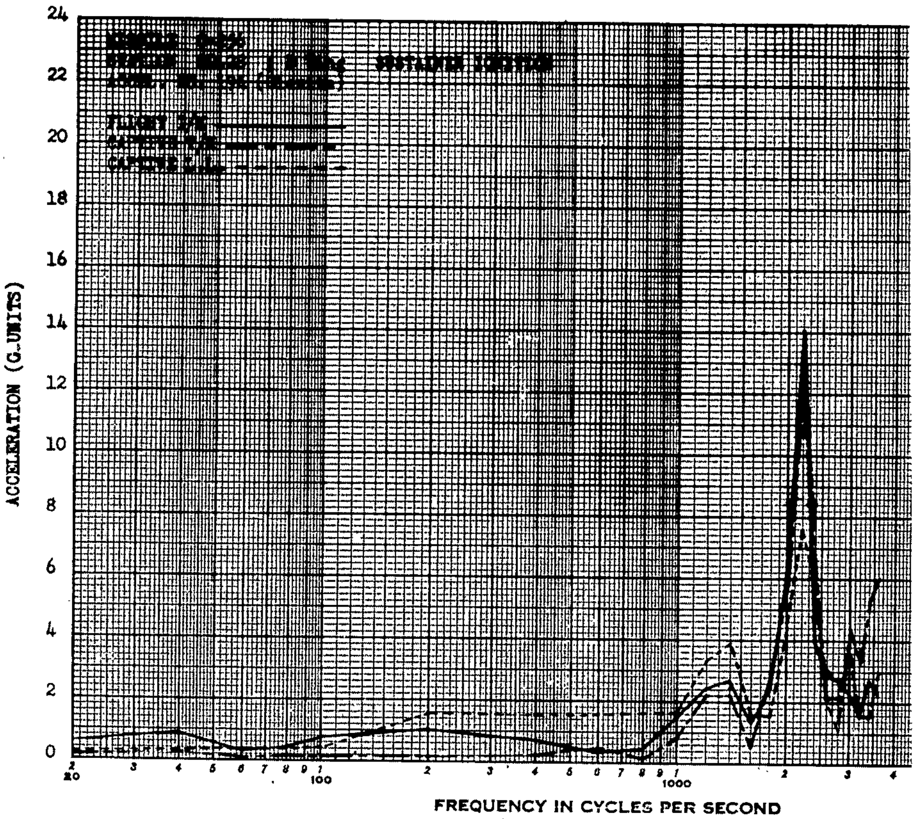


Figure 12

The pulse-height analyzer is a single-channel type manufactured by the Atomic Instrument Company. It has principal controls for adjusting "base line" and "channel width." If the channel width is set wide open, the instrument produces an output pulse whenever an input pulse exceeds the reference voltage set by the base-line control. If the channel width is set at one volt, for example, the output contains only those pulses whose peaks lie between the base line and one volt above the base line. The channel width control thus determines the amplitude range of pulses passed by the instrument. The analyzer has been found to perform satisfactorily with input pulses from 1 to 100 volts, but it does not give an accurate count of pulses less than 1 volt in amplitude.

METHOD OF OPERATION

The amplitude distribution was obtained by playing the recorded signal repeatedly, increasing the analyzer base-line voltage by increments equal to the channel width, and recording the count of the output pulses for each setting. The peak-sampling device reached saturation at about 24 volts pulse amplitude, hence, the playback gain was adjusted so that the maximum peaks in the signal fall below this limit. The pulse amplitudes in volts were converted to acceleration levels in g's by comparison with a calibration signal which was included on the same tape with the vibration record.

Table 3 is a typical set of data obtained with this technique. It will be noted that the number of pulses in the range 0-1 volt was found in a different manner, which was necessary because the analyzer was not accurate below the level of 1 volt. The maximum possible count was assumed to be the center frequency of the filter multiplied by the time length of the sample. The total number of pulses that exceeded 1-volt amplitude were measured with the analyzer by setting the base line at 1 volt and the channel width at maximum. The number of pulses less than 1 volt were then the difference between maximum count and the number exceeding 1 volt.

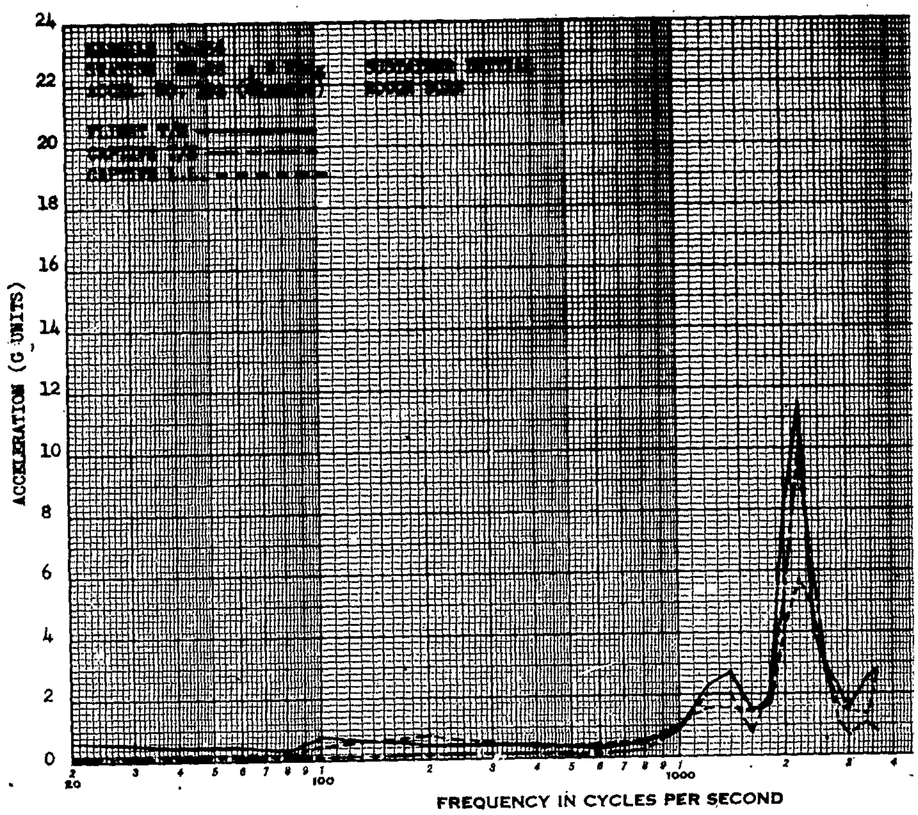


Figure 13

TABLE 3
Typical Amplitude Analysis

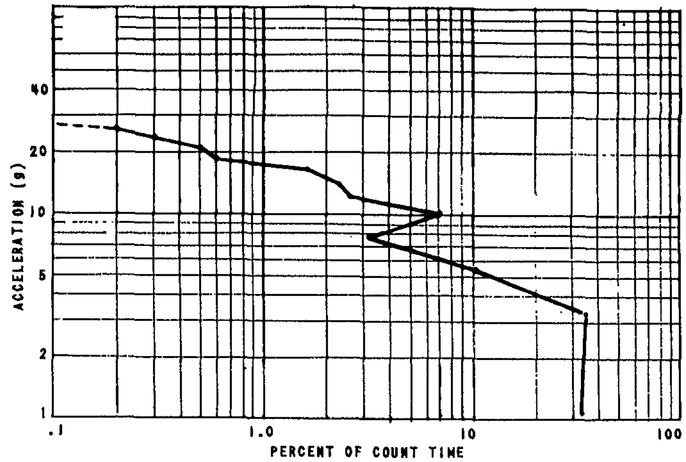
Baseline volts	Acceleration level (g)	Amplitude range	Count	Per-cent
1	3.3	1-2	425	34.5
2	5.5	2-3	127	10.3
3	7.8	3-4	40	3.2
4	10.0	4-5	85	7.0
5	12.2	5-6	33	2.7
6	14.4	6-7	27	2.2
7	16.6	7-8	20	1.6
8	18.8	8-9	8	.6
9	21.0	9-10	6	.5
10	23.3	10-11	4	.3
11	25.5	11-12	2	.2
12	27.7	12-13	0	----

Filter: 400-600 cps; Length of sample: 2.5 sec.; Maximum count: 2.5 x 490 = 1,225; Count exceeding 1 volt: 825; Count 0-1 volt: 400; Percentage 0-1 volt: 32.5; Calibration 1 V = 2.22 g.

The pulse count in each amplitude range divided by the maximum count gave a ratio or percentage which was a measure of the relative count for the corresponding acceleration level.

The data obtained from the amplitude analysis were plotted in different ways, two of which are illustrated in Figure 14. These charts portray the time distribution of acceleration levels for the frequency band of the filter used. They provide a statistical or probabilistic representation of the vibration environment. The root-mean-square (rms) acceleration amplitude, predominant amplitude, and other characteristic values may be computed. Acceleration spectral density curves may be obtained by plotting the results of a series of such measurements at different frequencies.

With these data, it should be possible to realize a three-dimensional display which takes into consideration the duration of a vibration, as mentioned earlier in this report. The approach to this problem has been to plot a family of spectral distribution curves as shown in



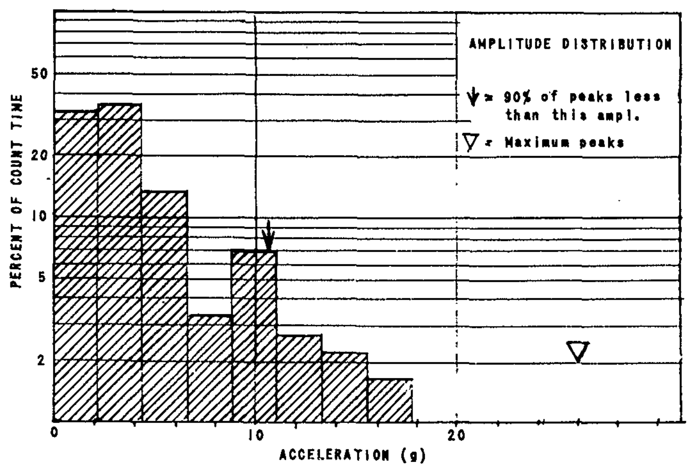


Figure 14 - Two ways of plotting amplitude distribution data

Figure 15. Each point plotted in the second curve is such that 90 percent of the peaks in the vibration waveform are less than this amplitude. These points are obtained by adjusting the pulse-height analyzer to a level such that a count of 10 percent of the maximum is measured and the other curves are obtained in an analogous way. The "less than" curves summarize the information gained from a series of measurements on a given vibration sample.

The experimental equipment used in preliminary evaluation of the system did not include the array of filters that were used to obtain the Miller oscillograph records mentioned previously. The electronic filters that were used had a much wider bandpass, especially at high frequencies. Figure 16 shows a comparison of the curves obtained from the same sample with the wide-band electronic filter and with a sharp 200-cycle bandpass filter. It is apparent that the principal peaks may be recognized using either bandpass. Some detail is, of course, lost in the wide-band case.

Various modifications of the above-described method are possible. For example, a multi-channel pulse-height analyzer could be used to obtain a series of points on the distribution curve with one playing of the sample. Such instruments, with ten or more channels, are commercially available; these are usually equipped with a separate counter or integrator for each channel. A further refinement might be a sequential sampling device which would sample the various integrator outputs and provide an ink recording of the amplitude distribution.

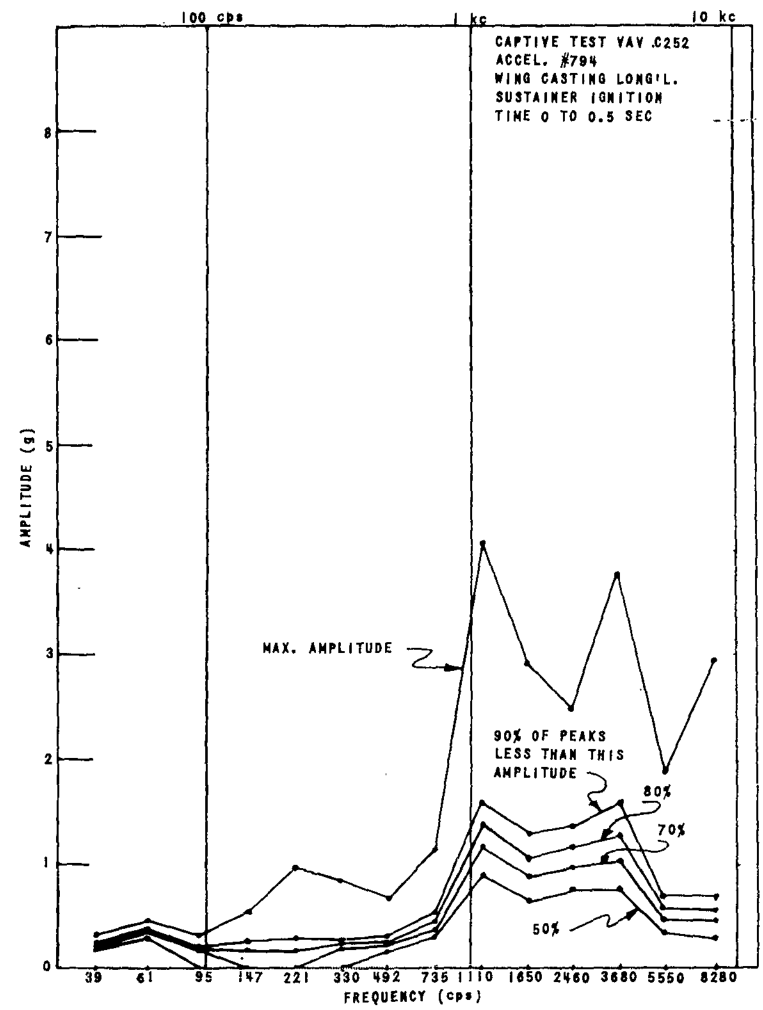


Figure 15 - Spectral distribution curves

DESCRIPTION OF PEAK SAMPLING DEVICE

The instrument that converts the complex input signal into a series of pulses, suitable for analysis in a pulse-height analyzer, is designated as the Peak Sampling Device. The circuitry used to perform this operation is outlined in the block diagram, Figure 17, and the complete schematic diagram is shown in Figure 18.

At every positive peak of the input signal, the peak-holding capacitor is charged to the peak voltage through a silicon diode (IN210), which allows the charge to remain for a time after the peak is passed. A sampling pulse, generated by Pulse 1 Multivibrator (MVR), drives the grid of the output cathode follower positive until it matches the voltage on the capacitor. Thus an output pulse is formed which is equal in amplitude to the peak voltage of the input, except for the slight loss of gain in the cathode followers. Following the sampling pulse, a discharge pulse is generated by Pulse 2 Multivibrator. This pulse causes the discharge tube (V10A) to conduct, discharging the circuit so that the sequence begins again with the next positive peak.

The grid of the peak detector (V2A) is connected to a capacitor which is charged through a diode to a little below the peak of the input signal. When the cathode of the peak detector passes a peak and starts back in a negative direction, the grid capacitor remains charged because of the diode coupling, thus causing a

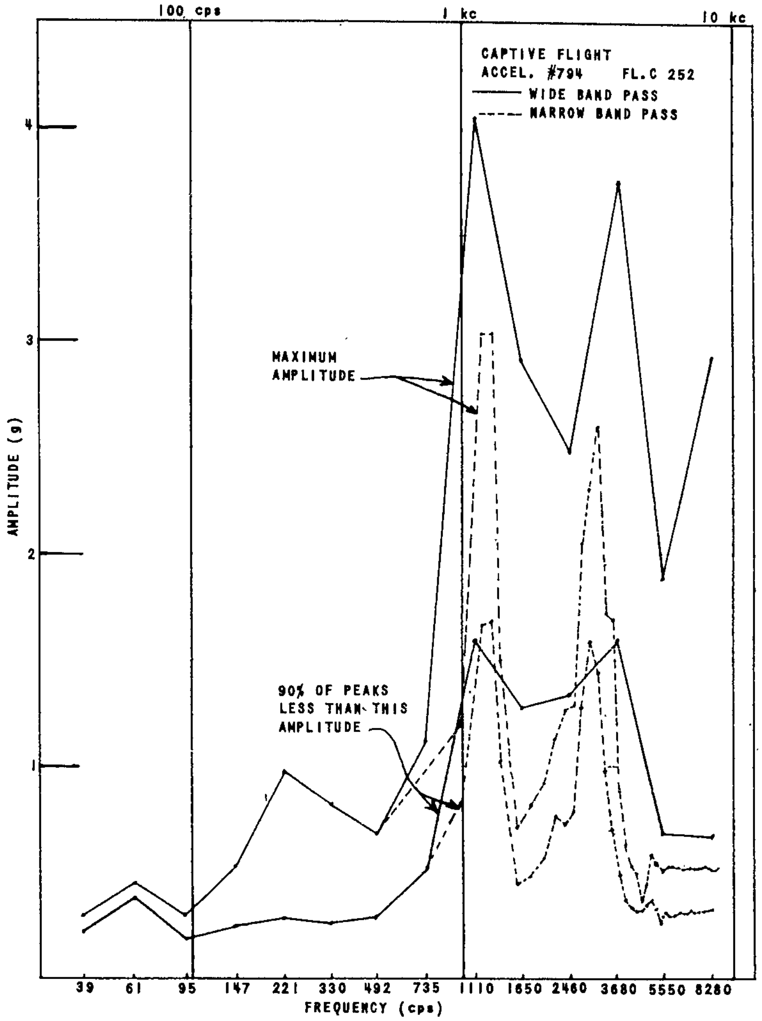


Figure 16 - Comparison of wide and narrow bandpass

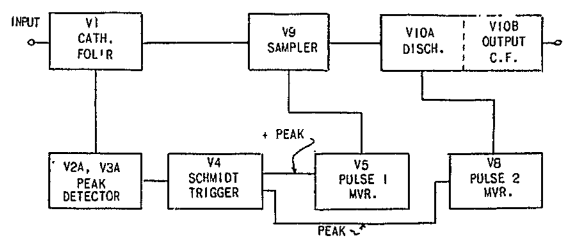


Figure 17 - Peak sampling device, block diagram

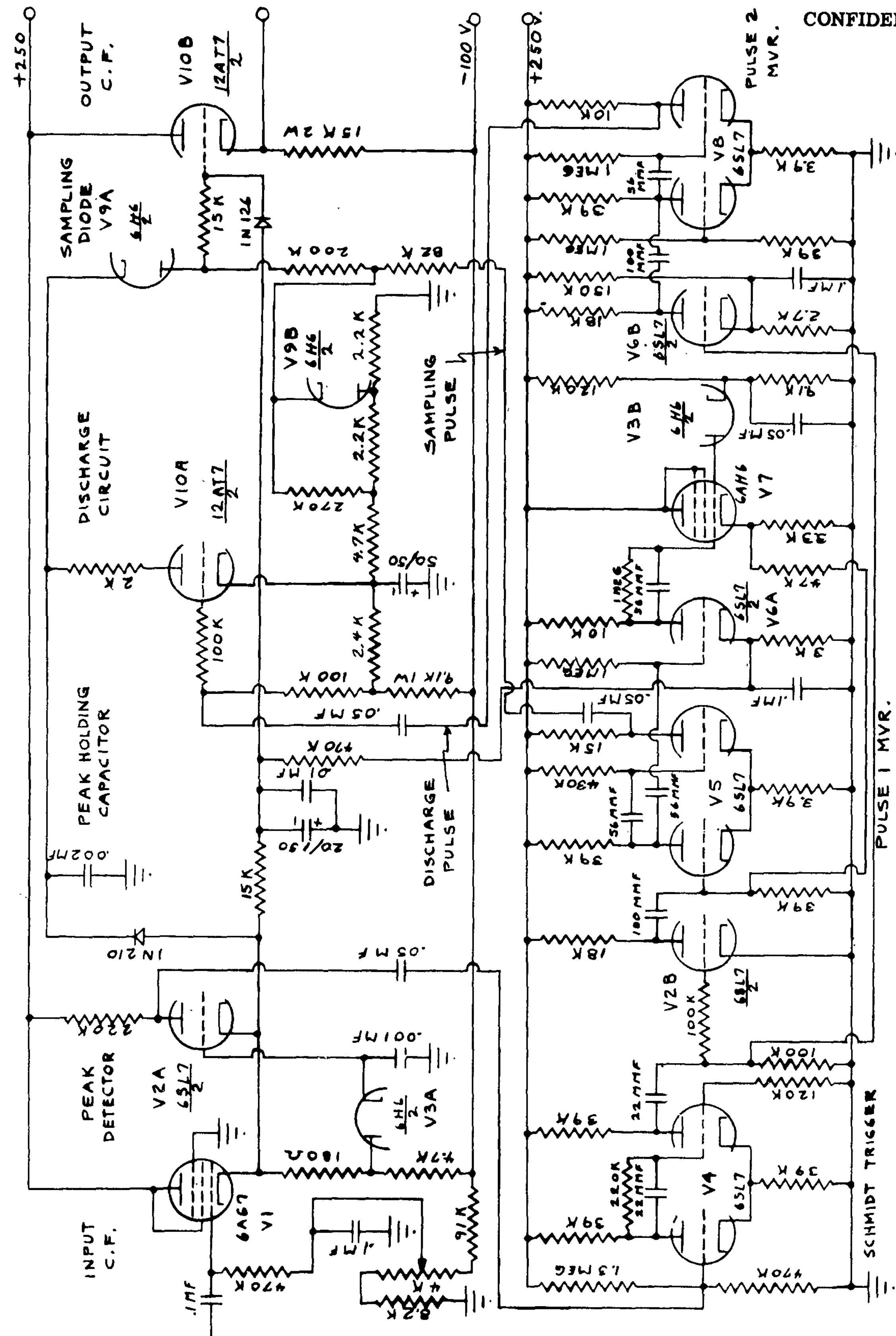


Figure 18 - Schematic wiring diagram of peak sampling device

sharp increase in plate current. The tube continues to conduct until the signal reaches a negative peak and again changes direction. The peak detector plate drives a Schmidt trigger circuit (V4) which produces a sharp negative swing at each positive peak and a positive swing at each negative peak. This output is differentiated and used to trigger the sampling and discharge pulses in the proper sequence. The Schmidt circuit provides constant amplitude trigger pulses independent of their frequency and the input signal amplitude.

Pulses 1 and 2 Multivibrators are fairly standard "one-shot" type circuits. Pulse 1 Multivibrator has a special circuit added to it which prevents "firing" more than once on one trigger pulse. The 6AH6 cathode follower (V7) normally holds the Multivibrator input grid (V5) at about plus 9 volts, so that an incoming positive trigger pulse is accepted. Once the Multivibrator fires, the grid of the 6AH6 is driven negative holding the input tube cutoff for a period of eight to ten microseconds.

By means of the potentiometer in the grid return of the input cathode follower, the dc level of subsequent circuits is adjusted so that the output pulses start from ground potential. The IN126 crystal diode clamps the grid of the output cathode follower to the mean dc level at the cathode of the input tube, the ac components being filtered out by an RC filter consisting of a 15,000-ohm resistor and 20-microfarad capacitor.

Figure 19 shows the results of a series of measurements made with the peak sampling device. The circled points indicate the spread of measurements made at different frequencies ranging from 100 to 20,000 cps. The linearity is excellent up to 24-volts output although saturation occurs slightly above this figure.

The ability of the device to follow rapidly changing peak amplitudes is illustrated in Figure 20. The maximum amplitude in this figure is about 15 volts. The signal has a beat modulation obtained by combining frequencies of approximately 75 and 100 cps.

LIMITATIONS AND ACCURACY

The limitations and accuracies of each component used in the shock and vibration determinations have not been defined in this paper

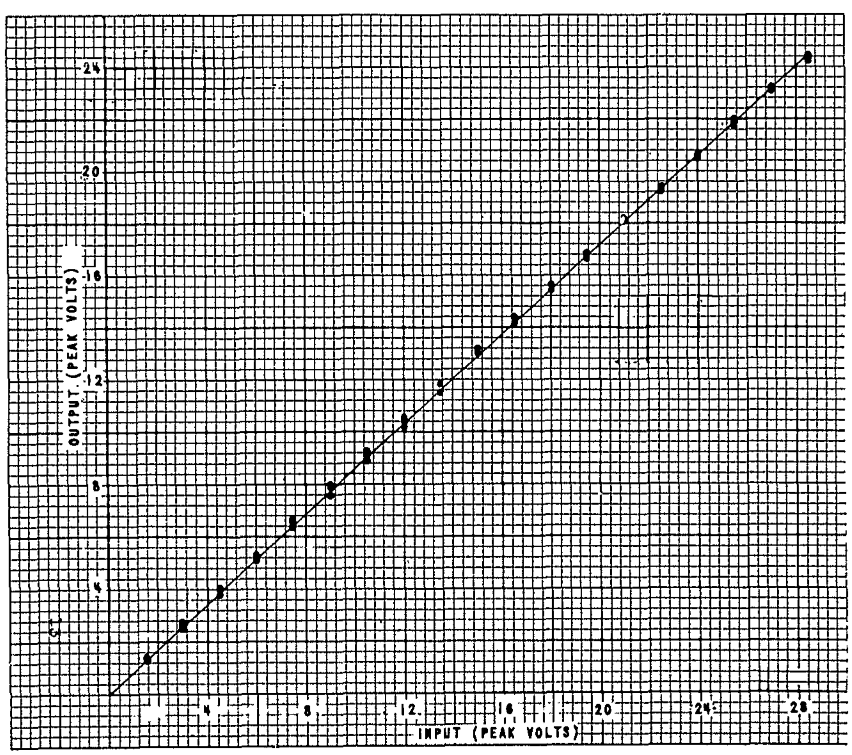


Figure 19 - Output pulse vs. input frequencies 100 cps - 20 kc

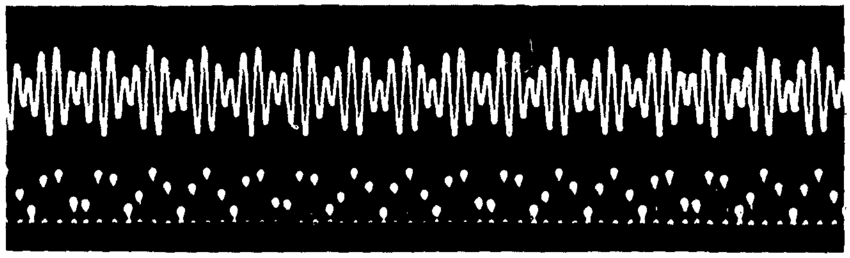


Figure 20 - Input and output of peak sampling device

because of constant changes occurring in the pickup instrumentation and combinations with associated circuit equipment. However, by weighing each segment in the environmental determination, the accuracy of the results can be said to be 60 percent or better.

CONCLUSIONS

1. The graphical presentations of vibration information illustrated in Figures 11, 12 and 13 confirm that the results obtained by captive landline and captive telemeter are essentially the same.

2. Figures 11, 12 and 13 illustrate that there is a high degree of correlation between the captive-test information and the free-flight information within the frequency range of the instrumentation.

3. The digital and analogue methods of reduction when employing the same band-pass filter give comparable results.

4. The digital method when employing wide bandpass filters and electronic readout equipment can shorten the time of reduction and analysis of a flight from weeks to days.

5. The analogue method when employed, yields the complete vibration picture, relative to time and orientation, within the limitations of the instrumentation.

6. The peak sampling device used in the digital method is noise free and produces uniform pulses. These are advantageous over certain other sampling methods which have been used for amplitude analysis.

7. The analogue and digital methods are adaptable to an automatic adjustment for correction of the frequency response of the instrumentation system.

8. The Miller oscillograph records of the analogue methods give a vivid visual picture of the shock and vibration with respect to time and orientation, see Figure 7.

9. The "less than" curves should be useful in the determination of specifications for missile components.

RECOMMENDATIONS

1. That there should be standard procedures for the calibration of accelerometers and associated circuitry.

2. That investigations be instituted to extend the usable frequency range of telemetering and other instrumentation concerned in vibration determinations.

3. That the designers and laboratory test facilities standardize on vibration information presentation required for their use.

4. That captive tests be continued for all missiles.

5. That further automation of both methods be investigated.

DISCUSSION

B. Baker, Douglas Aircraft: Mr. Fine, I would like to know whether there was only one or were there many captive firings made in order to get a statistical distribution of the various frequencies and g levels and to see if the actual flight data fell within the range?

Fine: More than one firing was made with a particular missile. The first time it happened it was accidental, but the results were satisfying. The second time the results were incidental because we were trying to evaluate the method to find out if the system was satisfactory or perhaps one which would only give us more completely confusing information. Actually we felt that the correlation was satisfactory. Not 100 or 80 percent but more like 50 or 60 percent. Of course they were not the same.

You cannot substitute captive testing for flight because your environment is not the same. As far as the sustainer motor burning was concerned, it was the same. But with aerodynamic forces acting on the missile, the picture does change. You get shocks transmitted through the missile when the sustainer motor burns out and every time the missile makes a drastic turn. You also get shocks when a booster separates. A program is now going on to determine the booster shock.

Bush, White Sands Proving Ground: What frequency response do you conscientiously feel you achieved throughout?

Fine: The NAMTC at Point Mugu calibrated the Endevco accelerometers for us and were present when the tests were recorded. They calibrated through 4,000 cps and therefore we are certain of frequencies to this level. Above 4,000 cps I believe the curves were extrapolated or comparisons were made with others. Now this was all land-line information. The subcarrier of the telemeter is on 70 kc and according to all the figures we should only get a frequency range of about 2.2 kc, although I believe we did a little better. But we have high hopes for the information which we have extrapolated.

* * *

SIMULATION OF ROCKET MOTOR VIBRATION BY STATIC TESTING

T. A. Angelus, Allegany Ballistics Laboratory, Hercules Powder Company
and D. A. Stuart, Cornell University

Disturbances occurring in rocket motor thrusts are analyzed both theoretically and experimentally. The theoretical analysis consists of formulation and solution of the general equations of motion for multi-degree-of-freedom systems. The experimental analysis consists of the accumulation of experimental data from static firings. A comparison between experimental and computed data is made and discussed.

INTRODUCTION

This paper results from investigations of shock and vibration carried out at the Allegany Ballistics Laboratory with cooperation from personnel at Wright Air Development Center and Northrop Aircraft.

Since disturbances occurring in steady state thrust are of concern to all in the guided missile program, an attempt was made at this laboratory to analyze one of the more complex booster systems. The problem at hand was to identify oscillations on thrust records as due to either real internal disturbances resulting from the ballistic properties of the propellant or mechanical amplifications of white noise not characteristic of the propellant.

To simulate the booster-missile attachments, a special thrust mount was utilized. This thrust mount was constructed so as to have the highest possible frequency response in order to measure the vibrational characteristics of "steady" thrust. The system was then analyzed making use of Newton's Second Law of Motion, Rayleigh's Method, and Lagrange's Equations in the solutions of problems containing multi-degrees-of-freedom.

It was shown from this investigation that the high-amplitude "noise" appearing on the thrust-time records of this unit was, for the most past, mechanical amplification of white noise. Experimental data to support this analysis are presented.

THE EQUATION OF MOTION

Rigid Body Analysis

Consider a canted-nozzle rocket motor mounted as in Figure 1. This simulates the missile system with strain gauges at the main thrust pad (k_1) and on the forward (k_3) and aft (k_2) sway braces. The signal as received from the thrust gauges is assumed to be a sinusoidal function of time. Neglecting damping in the analysis, the equations of motion of the motor for the three degrees of freedom considered are (spring mass being considered negligible):

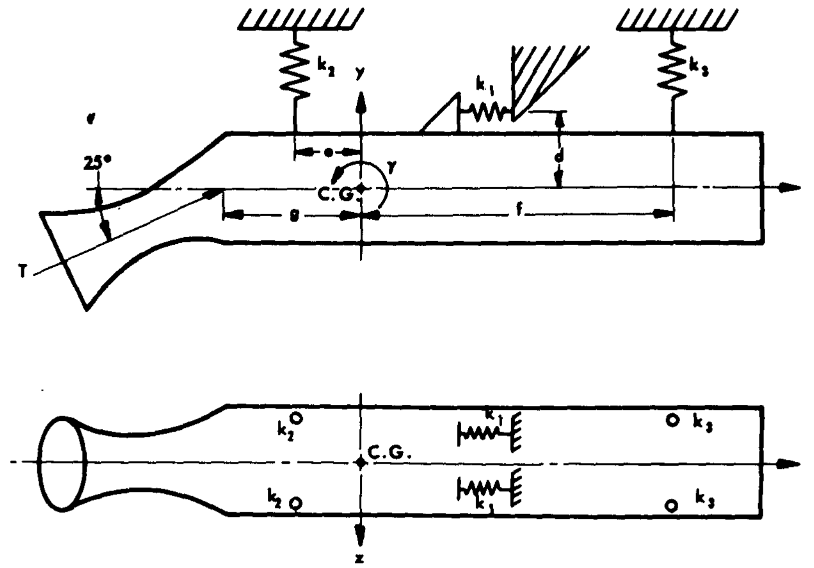


Figure 1 - Sketch of thrust mount

k_1 = gauge constants γ = rotation in xy plane

T = thrust   C.G. = center of gravity

$$m\frac{d^2x}{dt^2} + 2k_1x - 2k_1d\gamma = T\cos 25°,$$

$$m\frac{d^2y}{dt^2} + 2(k_2+k_3)y - 2(k_2e - k_3f)\gamma = T\sin 25°, \quad (1)$$

$$I_z\frac{d^2\gamma}{dt^2} - 2k_1dx - 2(k_2e - k_3f)y$$

$$+ 2(k_1d^2 + k_2e^2 + k_3f^2)\gamma = -Tg\sin 25°$$

where

m = mass of motor,

x, y, z = rectangular co-ordinates,

γ = rotation in xy plane,

T = thrust,

d, e, f, g = distances from centerlines,

k_i = measured gauge constants of proportionality,

I_z = moment of inertia,

t = time.

Rotations in the xz and yz planes were neglected for simplicity, because such motions were not sensed experimentally and because the coupling of such rotations with the three remaining displacements (x, y, and γ) was small.

To determine frequency response, we assume

$$T = A\sin\omega t,$$
$$x = A_1\sin\omega t,$$
$$y = A_2\sin\omega t, \quad (2)$$
$$\gamma = A_3\sin\omega t$$

where

$A_{1,2,3}$ = maximum amplitude,

ω = circular frequency.

By substituting these values (2) into equation (1) and dividing by $A\sin\omega t$, we have

$$-m\omega^2\frac{A_1}{A} + 2k_1\frac{A_1}{A} - 2k_1d\frac{A_3}{A} = \cos 25°$$

$$-m\omega^2\frac{A_2}{A} + 2(k_2+k_3)\frac{A_2}{A} - 2(k_2e - k_3f)\frac{A_3}{A} = \sin 25° \quad (3)$$

$$-I_z\omega^2\frac{A_3}{A} - 2k_1d\frac{A_1}{A} - 2(k_2e - k_3f)\frac{A_2}{A}$$

$$+ 2(k_1d^2 + k_2e^2 + k_3f^2)\frac{A_3}{A} = -g\sin 25°.$$

Solving for amplitude ratio A_1/A, by means of Cramer's Rule we get

$$\frac{A_1}{A} = \frac{\begin{vmatrix} \cos 25^\circ & 0 & -2k_1 d \\ \sin 25^\circ & 2(k_2 + k_3) - m\omega^2 & -2(k_2 e - k_3 f) \\ -g \sin 25^\circ & -2(k_2 e - k_3 f) & 2(k_1 d^2 + k_2 e^2 + k_3 f^2) - I_z \omega^2 \end{vmatrix}}{\Delta} \tag{4}$$

where

$$\Delta = \begin{vmatrix} 2k_1 - m\omega^2 & 0 & -2k_1 d \\ 0 & 2(k_2 + k_3) - m\omega^2 & -2(k_2 e - k_3 f) \\ -2k_1 d & -2(k_2 e - k_3 f) & 2(k_1 d^2 + k_2 e^2 + k_3 f^2) - I_z \omega^2 \end{vmatrix}$$

Solutions for A_2/A and A_3/A are done in a similar manner.

Those values of ω for which $\Delta = 0$, are the natural, or resonant frequencies of the system. To find the resonant frequencies, it is merely a matter of finding the roots to this frequency equation ($\Delta = 0$) which, when expanded, gives a sixth degree equation. Solving for ω^2 by means of Graeffe's method, resonant frequencies were determined for the system. The following two conditions of the motor were considered: (1) loaded and (2) expended.

The gauges do not measure the displacement x, y, and γ directly. The signals from the gauges are proportional to $x + f\gamma$, $y - e\gamma$, and $x - d\gamma$ for the forward, aft, and main gauges respectively. Since interest is in the amplitudes of the corrected gauge signals, we plot as functions of frequency:

$k_3 \frac{A_2}{A} + k_3 f \frac{A_3}{A}$ for the forward gauge,

$k_2 \frac{A_2}{A} - k_3 e \frac{A_3}{A}$ for the aft gauge, and (5)

$k_1 \frac{A_1}{A} - k_1 d \frac{A_3}{A}$ for the longitudinal thrust gauge.

Using the physical constants of construction, calculations for amplification factors using random frequencies in equation (5) and for resonant frequencies in the denominator of (4) were carried out and plotted into resonance diagrams for the various gauges. The resonance curves are shown in Figures 2, 3, and 4.

In this formulation, viscous and dry friction were neglected, thereby making the values for the maximum amplification factor frequencies slightly higher than what would be expected. Further, the neglect of damping yields infinite amplification factors. The actual amplification factors are, of course, finite.

Elastic Body Analysis

Since the previous equations considered the motor as a rigid body, it was found necessary in the course of this investigation to analyze the system as though it were vibrating in an elastic condition.

It was considered that the motor was vibrating similar to an elastic beam with spring suspensions. In an effort to find the fundamental frequencies of the system vibrating in this fashion, a comparative example was used. The following equations will give only an approximation of the fundamental frequency due to the simplifying assumptions made.

Consider the motor as a uniform elastic beam suspended by springs as illustrated in Figure 5. This is an infinite-degree-of-freedom problem and is handled briefly in two steps; being, (1) reduction of the problem to a finite number of degrees of freedom by the method of Rayleigh and (2) formulation of the equations of motion of the system with the application of Lagrange's Equations whose general form is

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{q}_s}\right) - \frac{\partial L}{\partial q_s} = 0 \tag{6}$$

where q_s is some generalized co-ordinate, and $L = T - V$, the Kinetic Potential (3).

Assuming the spring constant $k = (k_2+k_3)/2$, and that the beam has some initial deflected shape, we have, neglecting gravitational forces and using half the beam energy,

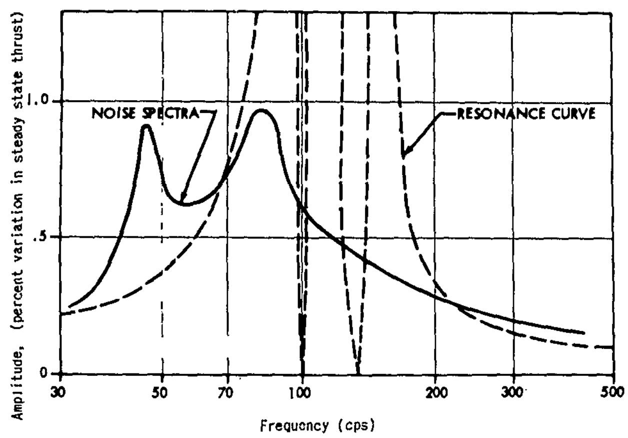


Figure 2 - Comparison of experimental noise spectra with computed resonance curves (forward gauge)

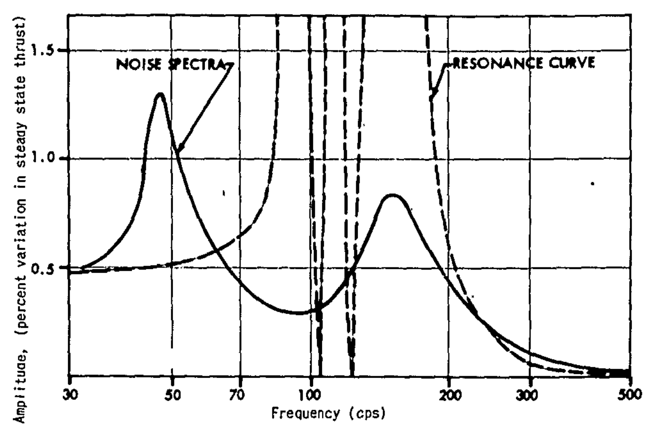


Figure 3 - Comparison of experimental noise spectra with computed resonance curves (aft gauge)

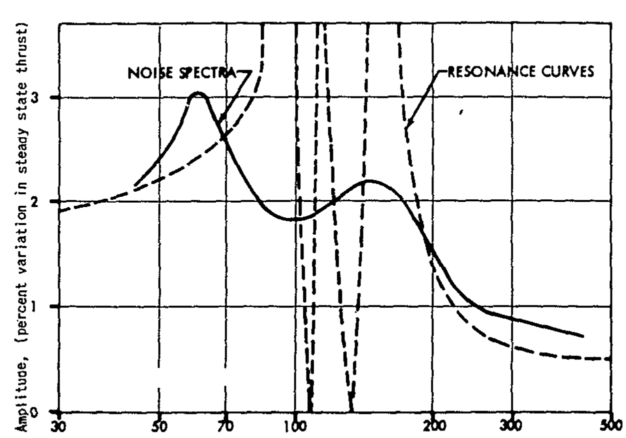


Figure 4 - Comparison of experimental noise spectra with computed resonance curves (main gauge)

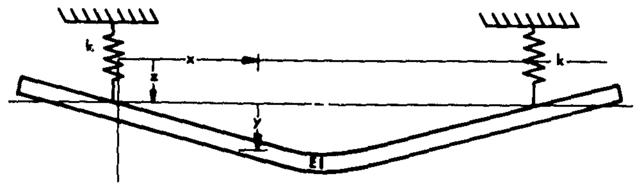


Figure 5 - Calculation of potential and kinetic energy of an elastic beam suspended by spring
z = deflection of spring $\quad$ Y = deflection of beam
k = spring constants $\quad$ EI = beam stiffness

$$V = \text{Potential Energy} = \frac{kZ^2}{2} + \frac{EI}{2}\int_o^L \left(\frac{\partial^2 y}{\partial x^2}\right) dx,$$
$$T = \text{Kinetic Energy} = \frac{\mu}{2}\int_o^L (\dot{Z} + \dot{y})^2\, dx \qquad (7)$$

where

μ = linear density,

Z = deflection of the spring,

y = deflection of the beam at any point x measured with respect to the end deflection as shown in Figure 2.

Assuming symmetry and harmonic motion, we have

$$y = A \sin\frac{\pi X}{2L} = a \sin \omega t \sin\frac{\pi X}{2L},$$
$$Z = b \sin \omega t. \qquad (8)$$

Using as effective co-ordinates Z and A, the Lagrangian forms of the differential equations of motion are

$$\frac{\partial V}{\partial A} = -\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{A}}\right), \quad \frac{\partial V}{\partial Z} = -\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{Z}}\right). \qquad (9)$$

Substituting (8) in equation (7) and applying (9), we get for the frequency equation:

$$\begin{vmatrix} \frac{k}{\mu L} - \omega^2 & \frac{-2\omega^2}{\pi} \\ \frac{-4\omega^2}{\pi} & \frac{EI}{\mu}\left(\frac{\pi}{2L}\right)^4 - \omega^2 \end{vmatrix} = 0. \qquad (10)$$

Again using the physical constants of construction in equation (10), we get for the two modes of vibration:

$$f_1 = 38.2 \text{ cps}, \quad f_2 = 264 \text{ cps}.$$

By using the simple relations:

$$\omega^2 = \frac{EI}{\mu}\left(\frac{\pi}{2L}\right)^4 = \text{Frequency of beam with pin ends},$$

$$\omega^2 = \frac{k}{\mu L} \quad = \text{Frequency of mass on springs}.$$

it is found that their respective frequencies are

$$f_1 = 40.4 \text{ cps}, \quad f_2 = 109 \text{ cps}.$$

Thus in either of the above cases, a large amplification would be expected experimentally in the vicinity of 40 cps. It is therefore expected that the four-degree-of-freedom system would also yield a large amplification factor at about the same frequency.

EXPERIMENTAL RESULTS AND DISCUSSION

Figure 6 is a three dimensional sketch of the unit depicting the location of the strain-type gauges utilized. As shown in Figure 7, these gauges were constructed of steel with a flat midsection and with holes at each end. The gauge ends were attached to the motor and to the firing bay floor with clevis pins. A strain grid which was attached to the flat midsection of each gauge formed an active arm of a conventional Wheatstone bridge. A steel block to which was attached a strain grid, was positioned beside each active strain gauge. This assembly acted as a temperature compensator and became a second arm of the Wheatstone bridge. Two dummy arms (strain grids without steel bases) were included to complete the bridge setup. The output from the bridge was fed to an inverted amplifier and, thence, to the oscilloscope where pictures were taken on a rotating drum camera.

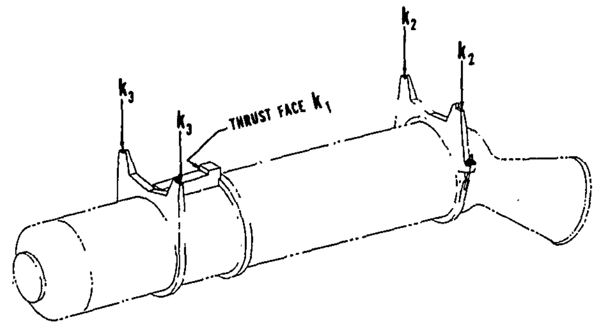


Figure 6 - Sketch of motor showing mounting structure and location of gauges

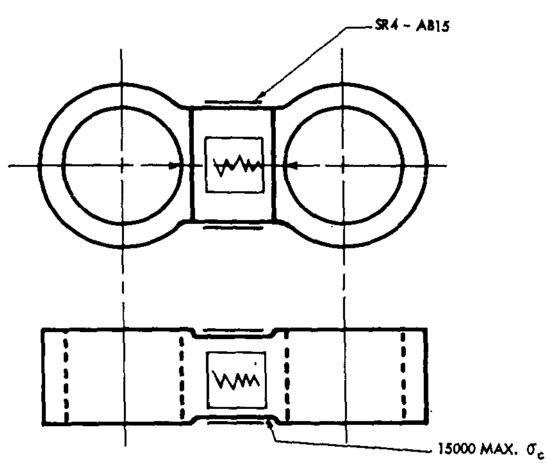


Figure 7 - Basic design of gauges

The testing program was planned to include firing tests of units conditioned at both high and low temperatures. One of the drawbacks of the above setup involved difficult alignment problems prior to firing during which time the unit underwent appreciable temperature change. Others were non-linearities induced by bending of the pins and gauges themselves due to loads imposed during firing.

The methods used in determining frequencies and amplitude of the vibrations occurring on the thrust-time records were relatively simple, and extreme accuracy of measurement is not claimed. The method consisted of measuring relatively pure vibration from an extended thrust-time record. The values of amplitude were then plotted semi-logarithmically as functions of frequency, giving representative noise spectra of the system for the various gauges.

By comparing the experimental noise spectra with the computed resonance diagrams it was assumed possible to isolate the resulting amplitude peaks into two cases being, (1) real internal disturbances resulting from the ballistic properties of the propellant or (2) mechanical amplifications of white noise, thus not characteristic of the propellant.

In Figures 3, 4 and 5, the following similarities can be noted. The amplitude peak at 150 cps on the aft and main gauges is apparently associated with the third mode of vibration. The disturbance at 85 cps on the forward gauge is probably associated with the first mode of vibration. The frequency at 45-50 cps, which is dominant on the forward and aft gauges is not associated with rigid body motion and, hence, does not appear in the analysis. Further investigation of the system as an elastic body showed that this disturbance, which occurred at the end of burning of the motor, resulted from the motor vibrating as an elastic beam.

The predominant amplitude peak at 65 cps on the main gauge was not determined from the limited analysis of the system. It is probable that this peak is due to a coalescence of a peak at approximately 40 cps, and one at approximately 100 cps due to damping and to the fact that all gauges sense a combination of translation (in either the x or y directions), bending, and rotations. Further quantitative work is suggested along these lines.

Since all peaks on the thrust records can apparently be attributed to mechanical amplification, it is concluded that an "actual" noise spectrum of the motion of constant amplitudes at all frequencies would not be inconsistent with the observed results. For this reason, it is suggested that this motor be characterized

as giving white noise with total amplitude of not more than 2 percent of full thrust.

SUMMARY

In this paper, an attempt was made to identify the disturbances occurring in steady-state thrust of a particular missile booster system. In order to simulate the actual environment of the missile, a special thrust mount was utilized for static firings. It might be mentioned that the k-values for the various gauges used in this study are not the same as those for the attachments incorporated on the missile in flight. The k-values on the flight missile are, in this case, of much lower magnitude. This, of course, means that the lowest resonant frequency of this system probably will be of lower magnitude.

In the formulations presented, damping was neglected. This, of course, is an oversimplification, since the percentage of damping is appreciable. If allowance is made for damping in comparison of data, the results appear reasonable.

The results from this investigation have shown that oscillations appearing on the thrust-time records of this particular unit can be attributed in the most part to mechanical amplifications arising from the natural motions of the system. To elaborate on this statement, it was shown by the comparison of computed resonance diagrams and noise spectra that disturbances which were by nature mechanical, could be identified. In most cases, this method was effective but in one other case, further analyses were necessary to determine the nature of the amplification. It is quite probable that all of the disturbances could be identified if a more thorough investigation were conducted. It is thus concluded that the actual "noise" output of the motor is probably white noise with an amplitude of ± 2 percent of mean thrust.

This method of identifying disturbances in thrust has also been carried out for many other systems and the conclusions have been similar in nature. It is recommended that similar analyses be considered on other rocket systems, when necessary, so that more effective methods can be taken in the correcting of conditions that are not within rocket specifications.

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* * *

STUDY OF METHODS
OF MEASURING ROCKET THRUST

Frank W. Bubb, Phillips Petroleum Company

This paper describes one of the thrust stands currently in use for measuring rocket thrusts. This stand is not properly designed for use with rockets having highly brisant ignitors. For such ignitors, the indicator card showing thrust-versus-time exhibits an initial high impulsive peak due to the ignitor. This is followed by a gap indicating zero thrust, and then the gap is followed by an excessively high impulsive peak, after which the thrust curve oscillates violently and approaches the steady thrust value somewhat as an exponentially attenuating sinusoid. Such curves are not true representations of rocket thrust, particularly during the early transient stage after firing--the transients being due to the faulty stand and not to the rocket. To those who, without understanding the role of the stand, have to make responsible judgments as to rocket performance, such cards are misleading--sometimes alarming. These phenomena are analyzed and explained in this paper. On the basis of this dynamical analysis, formulae are set up for the design of a shock absorber to relieve the initial thrust peak, and formulae for the design of a complete spring system for eliminating the gap, the excessive second peak, and the oscillatory approach to the steady thrust value. Application of the design formulae to the stand used for testing a specific Jato rocket is worked out and exhibited graphically.

Attention has been directed to the design of shock buffer springs for a stand currently in use for measuring the thrust produced by rockets with brisant ignitors. The unsatisfactory behavior of the stand without buffer springs has been studied, causes of its misbehavior investigated, and a redesign carried out.

DESCRIPTION OF THRUST STAND ASSEMBLY

The rocket is cradled in a channel iron and is fastened thereto by two chains wrapped around the upper half of the cylindrical rocket bottle. The channel is suspended by leaf springs as shown in Figure 1. At the forward or ignitor end of the channel, an adjustable bolt B threads into a steel block fixed to the channel. The bolt B may be turned and thereby adjusted longitudinally until its head comes into contact with the head H of the thrust measuring instrument. By further turning of bolt B, the thrust measuring instrument may be prestressed, in which case, the leaf springs and small helical spring S are deflected. A shaft from head H runs back into a cylindrical strain gage, the base of the container being fixed in a heavy concrete block. As the thrust T which B exerts upon H varies, so varies the resistance of the strain gage. This resistance is in one leg of a bridge circuit which yields a voltage proportional to the thrust T. This voltage is amplified and operates a cathode-ray tube, whose fluorescent spot then moves a distance proportional to T. The brilliant point on the cathode screen is focused upon a moving photographically sensitive screen, the

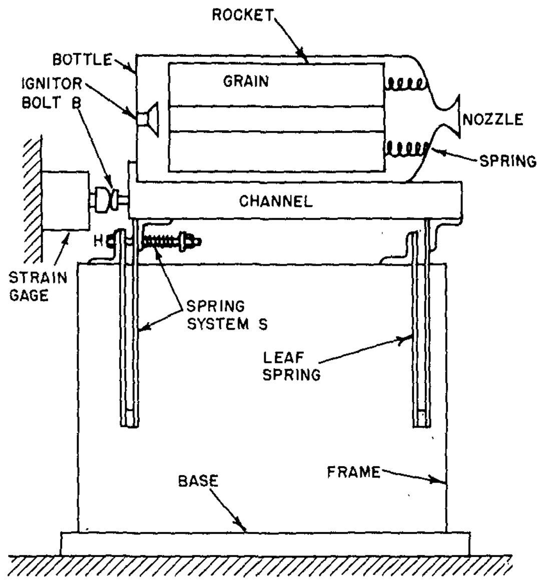


Figure 1 - Thrust stand assembly

motion of this paper being proportional to the time and at right angles to the motion of the bright point. In this manner, one obtains a graph of the thrust T versus the time t. A similar strain gage set-up actuated by the internal pressure P inside the rocket bottle yields a graph of the internal rocket pressure versus the time t.

The graphs of T versus t and P versus t obtained by this test stand and its instrumentation are typified by the curves shown in Figure 2. This figure shows only the early, transient and

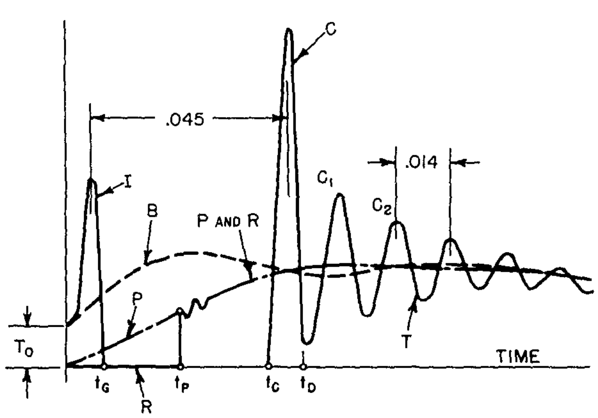


Figure 2 - Time graphs: P = pressure, R = reaction, T = thrust without buffer springs, B = thrust with buffer springs

violent part of the T graph—which approaches, of course, the graph of R, the actual rocket reaction.

PRINCIPAL FEATURES OF THRUST STAND PERFORMANCE

From the fluid mechanics of flow through the nozzle, it has been shown elsewhere that the rocket reaction R due to the supersonic jet is related to the pressure intensity P inside the bottle by the equation,

$$R = 1.48\ P.$$

Initially, there is a plastic plug in the nozzle whose purpose is to confine the gases from the burning fuel, allow the pressure to build up, and increase the rate of burning. The time when this plug blows out is shown on the P curve by a small jiggle at time marked t_p. During the time from 0 to t_p, the rise in P is shown by the graph. However, during this time, the rocket reaction R is zero, because the pressure in the closed bottle acts equally on both ends. After t_p, the equation R = 1.48 P applies. To keep the figure as simple as possible, the P and R graphs are shown as one after t_p - which may be done by supposing them to be plotted to different scales in the ratio 1.48.

It is important now to recall that the real purpose of this thrust stand and its instrumentation is to measure the actual rocket reaction R and its manner of varying with time. We note first that the force which we have denoted by T acts on the head H of the pressure transducer and is not at all the same force as R. Between the force R and the measured force T, there intervenes the dynamics of the stand. Our first task is, therefore, to account for the startling differences between the curves for T and R during the early or transient stage of the rocket firing. The T curve looks alarming and might well be misinterpreted as indicating an actual misbehavior of the rocket, whereas the P and hence the R curve are smooth except for the small bobble when the plug blows. This is shown, of course, by the actual experimental P curves. The experimental T curve settles down after the transients die out and approaches, of course, the experimental R curve. The alarming features of the T curve are explained as follows.

The first peak marked I on the T curve is due to the sudden reaction produced by the ignitor when it explodes. The ignitor fires its

charge toward the nozzle and ignites the rocket grain. The reaction impulse on the ignitor end of the bottle is transmitted through the channel and bolt B to the head H of the pressure transducer. The contact between B and H, which was severe because of the very brisant ignitor used, lasted only about 0.004 seconds, being broken at time t_G - this time $t_G = 0.004$ seconds being measured from an actual experimental graph. This indicates that contact between B and H acts like a very stiff spring. In fact, an estimate is easily made of the stiffness of this spring as follows.

The mass of the channel and rocket (270 lb) oscillates approximately sinusoidally as shown after time t_D. From measurements on an actual indicator card, the period of this oscillation was $\tau = 0.0136$ seconds. Using the well known formula from the theory of harmonic motion,

$$\tau = \sqrt{\frac{W}{k\,g}},$$

and inserting the numbers ($g = 386$), we get, after solving for k, the value $k = 149{,}000$ lb per inch spring rate.

A measure may also be made of the violence of the initial impulse imparted by the ignitor to the moving parts (rocket and channel). The area under the T versus t curve is a measure of momentum. The initial momentum of the moving mass is WV_o/g where $W = 270$ lb, $g = 386$ in per sec², and the initial momentum V_o is to be calculated.

Since this momentum was reduced by the B-H spring action to zero at the instant T attained its I peak maximum, we equate the area from $t = 0$ to this instant of maximum to the initial momentum WV_o/g. Using data from an experimental T versus t curve and solving for V_o, we get the value $V_o = 25$ in per sec. Had the spring action between B and H been softer, the high I peak would have been smaller and its time base t_G would have been much wider—thus eliminating the startling I peak. This indicates the introduction of a soft spring (with k much less than 149,000) between H and the moving mass.

At the time marked t_G, the bolt B obviously parts company with head H, since the instrument indicates a time gap of zero thrust thereafter. Since the moving parts are then going backward, a gap opens up between B and H. This gap was measured by fixing a pencil to the rigid frame of the stand so that its point bore upon a piece of paper fixed to the channel. The length of the mark on the paper, and consequently the length of the gap was of the order of 1/8 in.

The next event of significance is the plug blow out at time t_P. At this instant, the high pressure gases begin to flow out the nozzle at supersonic speed. A forward (to the left in Figure 1) reaction force R develops on the bottle, driving the channel to the left and closing the gap. While closing the gap, during which time no force opposes the jet reaction, the moving parts gain momentum. Consequently, when B comes again into contact with H, the head H suffers another impact - like that of a hammer striking an anvil.

The beginning of this impact occurs at time t_C. In stopping the moving parts, the thrust T rises rapidly and produces the peak marked C in Figure 2, the maximum of this C peak being from 3 to 5 times the actual jet reaction the T curve is intended to measure. In some cases, the rebound opens up the gap again and the T curve to the right of C goes down to the time axis as indicated by the extrapolated line crossing at t_D. The time interval from t_C to t_D is of the order of 0.008 sec—which indicates a still more powerful impact at C than the ignition impact I. The usual behavior, however, is shown in Figure 2 where T is exhibited as approaching the R curve like a damped sinusoid.

From this discussion, it should be clear that the gap (causing the astonishing C peak) is due to the fact that the S spring (see Figure 1) is too weak to hold bolt head B against transducer head H. This indicates that we should use a stiffer S spring suspension supporting the channel rocket cradle.

To sum up this section, the remedies for the whole dynamical misbehavior of the thrust stand consist in our adopting two measures: (1) introduce a buffer spring B between the channel cradle and the bolt head to take care of the ignitor impulse; and (2) use a stiffer spring S between the channel and the rigid base of the thrust stand to prevent the formation of a gap between B and H, which gap caused the extraordinary C peak.

THEORY OF THE BUFFER SPRINGS

We show schematically in Figure 3a the mass W, together with the B spring of stiffness K (lb per in) and the S spring of stiffness k. This figure illustrates the state where both springs

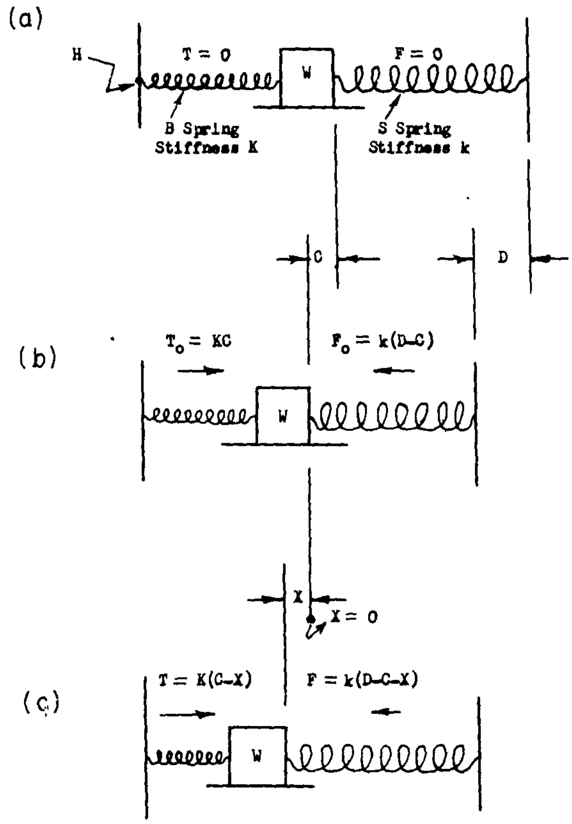


Figure 3 - Analysis of buffer spring action: (a) condition before pre-load, (b) condition after pre-load, (c) condition after firing W displaced X

just touch the mass W, with B and H just touching, and without exerting force on each other.

Figure 3b shows what happens when the spring S is compressed and a pre-load is applied. This pre-load is produced by adjusting the nut at the end of the helical part of the S spring. The nut end of S is moved a distance D, while the W end is moved a distance C, thus compressing the S spring the amount D-C and producing therein a compression k(D-C), which acts to the left on W. At the same time, the B spring is compressed C and exerts a force $T_o = KC$ on W to the right. T_o is the pre-load on the head H. When the mass W is in equilibrium,

$$T_o = KC = k(D - C) = F_o \qquad (1)$$

and the rocket is ready for firing.

Suppose now that the ignitor is exploded. This occurs in a very short time and imparts to W an initial velocity V_o to the left practically instantaneously, W moves to the left, thus increasing the compression in the B spring and reducing that in the S spring. When W has moved a distance x from its equilibrium pre-loaded position, the forces are as shown in Figure 3c—namely, the B spring exerts a force $T = K(C + x)$ to the right and the S spring exerts a force $k(D - C - x)$ to the left.

The resultant force on W to the right is

$$T - F = K(C + x) - k(D - C - x)$$
$$= (K + k)x - [KC - k(D - C)].$$

From (1), we see that the term in brackets is zero. Hence, the unbalanced force acting on W to the right is $G = T - F = (K + k)x$. This is the general expression for the unbalanced force on W for any value of x, whether x is positive or negative. Neglecting damping forces, the equation of motion of W is then,

$$\frac{W}{g}\ddot{x} = -(K + k)\,x. \qquad (2)$$

This differential equation is subject to the initial conditions,

$$\left.\begin{array}{l}\text{(1) When } t = 0, \text{ then } x = 0.\\ \text{(2) When } t = 0, \text{ then } \dot{x} = V_o.\end{array}\right\} \qquad (3)$$

The solution is well known and is

$$x = \frac{V_o}{\omega}\sin\omega t = x_m \sin\omega t \qquad (4)$$

where

$$\sqrt{\frac{(K + k)\,g}{W}} = \omega \qquad (5)$$

and the period of oscillation is

$$\tau = \frac{2\pi}{\omega}. \qquad (6)$$

The equation for the velocity at time t is

$$\dot{x} = V = V_o\cos\omega t = \omega x_m\cos\omega t. \qquad (7)$$

Since the thrust on head H is $T = K(C + x) = T_o + Kx$, we have

$$T = T_o + Kx = T_o + Kx_m\sin\omega t. \qquad (8)$$

Equations (4) (7) (8) show how the thrust stand behaves with buffer springs B and H in place during the time interval from $t = 0$ to $t = t_p$.

Suppose now that the nozzle plug blows out at time t_P and that the jet reaction R begins at this instant. To idealize a bit, let us assume that R is a step function, namely, that R increases instantly at time t_P from zero value preceding t_P to a constant value R for all time after t_P. Referring to Figure 3, we must now add a constant force R to the left on W in addition to the two spring forces. The new equilibrium position of W may be calculated from the equation,

$$R_o = KC' = R + k(D - C') \tag{9}$$

where R_o will be the steady thrust as measured by the pressure transducer. Solving (9) for C' and using (1), we get

$$C' = \frac{R + kD}{K + k} = \frac{R}{K + k} + C \tag{10}$$

which locates the new equilibrium position for W. Subtracting (see (1)) $C = kD/(K + k)$ from C', we locate the new equilibrium position of W at a distance

$$E = C' - C = \frac{R}{K + k} \tag{11}$$

to the left of the former equilibrium position. Using (1) and (10) in (9), we get

$$R_o = \frac{K}{K + k}(R + kD) = \frac{KR}{K + k} + T_o. \tag{12}$$

Solving (12) for R gives

$$R = \frac{K + k}{K}(R_o - T_o) \tag{13}$$

which is a useful formula since both R_o and T_o will be known.

But let us return now to the calculation of the T curve for time after t_P. Suppose that W be displaced a distance y to the left of its new equilibrium position specified by (11). The constant force R will now act to the left on W in addition to the spring forces. If one shows this force R in Figure 3, it is easy to see that the resultant force on W to the right is

$$G = K(C' + y) - R - k(D - C' - y)$$

$$= (K + k)y + [KC' - k(D - C') - R]$$

where the bracket is zero by virtue of (9). Hence, the resultant force is

$$G = (K + k)y \tag{14}$$

to the right on W. Note, however, that the thrust T on the pressure transducer head H is due only to the B spring (which is all that touches H) and has the value

$$T = K(C' + y). \tag{15}$$

Regarding y as positive to the left, the differential equation of motion of W is

$$\frac{W}{g}\ddot{y} = -(K + k)y. \tag{16}$$

This differential equation is of exactly the same form as (2), but is subject to the new conditions:

$$\begin{aligned} &(1)\ \text{When } t = t_P, \text{ then } y(t_P) = x_m \sin \omega t_P - \frac{R}{K + k} \\ &(2)\ \text{When } t = t_P, \text{ then } \dot{y}(t_P) = \omega x_m \cos \omega t_P. \end{aligned} \tag{17}$$

which mean respectively that the position of W and its velocity at time t_P must be the same whether calculated by using the x equations preceding t_P or the y equations which hold after t_P.

The solution of (16) is

$$y = y_m \sin \omega(t - t_o) \tag{18}$$

with

$$\dot{y} = \omega y_m \cos \omega(t - t_o). \tag{19}$$

The first condition in (17) requires that

$$y_m \sin \omega(t_P - t_o) = x_m \sin \omega t_P - \frac{R}{K + k} \tag{20}$$

while the second requires that

$$y_m \cos \omega(t_P - t_o) = x_m \cos \omega t_P. \tag{21}$$

Squaring these two equations and adding eliminates t_o and gives

$$y_m = \sqrt{x_m^2 - \frac{2x_m R \sin \omega t_P}{K + k} + \left(\frac{R}{K + k}\right)^2} \tag{22}$$

where, in using this equation to calculate y_m, we shall always take y_m to be positive, which can be done by proper adjustment of the phase t_o. After calculating y_m from (22), it is an easy matter to calculate t_o by use of trigonometric tables from (21).

Thus, $$t_o = t_P - \frac{1}{\omega} \arccos\left(\frac{x_m}{y_m} \cos \omega t_P\right). \tag{23}$$

Upon substituting the values for y_m and t_o given in (22) and (23) into (19), the motion after t_P is completely determined.

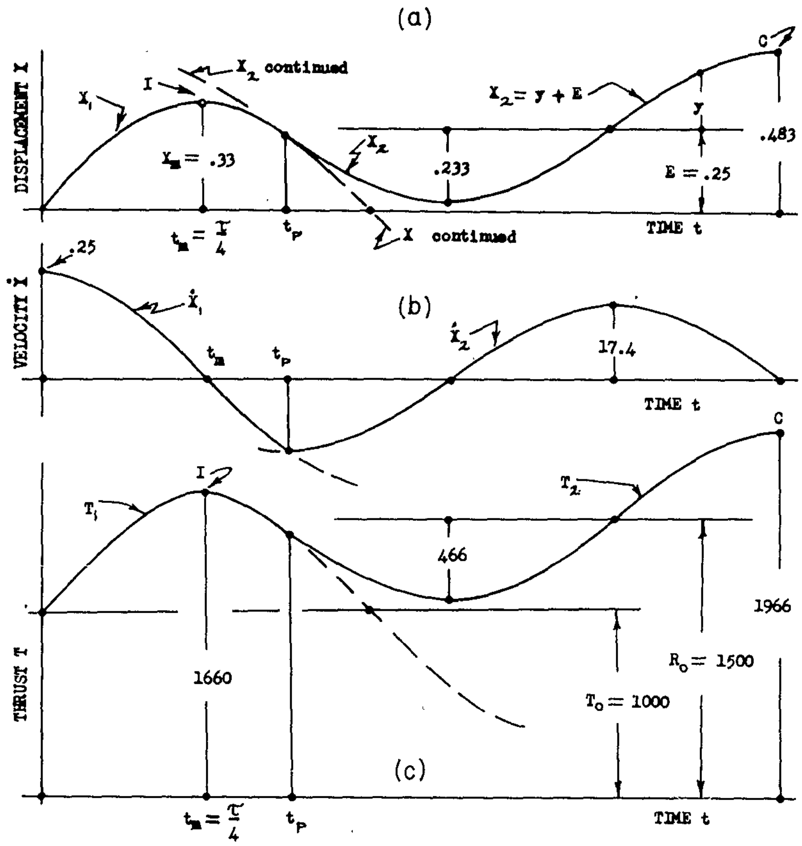


Figure 4 - Graphs of X, $\dot{X}$, T vs. t

Graphs showing how the two branches of the displacement curve (the x branch to the left of t_p and the y branch to the right of t_p) join at t_p are shown in Figure 4. Similar curves are shown for the velocity and thrust, where

$$T = R_o + Ky_m \sin \omega(t - t_o). \tag{24}$$

DESIGN EQUATIONS

For quick reference in design calculations, we summarize here the essential equations governing design of the B and S buffer springs.

The subscripts 1 and 2 will be used to distinguish quantities respectively preceding and following t_p in time. That part of a curve for $t < t_p$ will be called its Branch 1, while that part for $t > t_p$ will be called its Branch 2, t_p being the branch point.

BRANCH 1

$$x_1 = x_m \sin \omega t, \tag{A$_1$}$$

$$\dot{x}_1 = V_o \cos \omega t, \tag{B$_1$}$$

$$T_1 = T_o + Kx_m \sin \omega t, \tag{C$_1$}$$

BRANCH 2

$$x_2 = y_m \sin \omega(t - t_o) + \frac{R}{K + k}, \tag{A$_2$}$$

$$\dot{x}_2 = \omega y_m \cos \omega(t - t_o), \tag{B$_2$}$$

$$T_2 = R_o + Ky_m \sin \omega(t - t_o). \tag{C$_2$}$$

To make a tentative design, we assume the following parameters:

$$W,\ V_o,\ R,\ k,\ K,\ T_o,\ t_P, \tag{D}$$

and calculate the following quantities:

$$\omega = \sqrt{\frac{g(K - k)}{W}}, \quad \text{(E)}$$

$$\tau = \frac{2\pi}{\omega}, \quad \text{(F)}$$

$$x_m = \frac{V_o}{\omega}, \quad \text{(G)}$$

$$R_o = \frac{K\,R}{K + k} + T_o, \quad \text{(H)}$$

$$y_m = \sqrt{x_m^2 - \frac{2x_m R \sin \omega t_P}{K + k} + \left(\frac{R}{K + k}\right)^2}, \quad \text{(I)}$$

$$t_o = t_P - \frac{1}{\omega} \arccos\left[\frac{x_m}{y_m} \cos \omega t_P\right]. \quad \text{(J)}$$

Substitute these into (A) (B) (C), graph the performance curves. If the performance curves are not satisfactory, one assumes a different set of values for the spring stiffnesses k and K, and tries again.

Concerning the parameters (D), the following values are the final ones: W = 270 lb (given), V_o = 25 in per sec (average value for the present ignitor—although efforts are being made to reduce this), R = 1,000 lb (thrust for which this jato was designed—although this will be measured in every performance test), T_o = 1,000 lb (selected after a few trials as satisfactory), t_P = .031 seconds (average of many experimental values), k = K = 2,000 lb per in (selected as satisfactory after a number of trials). Using these values in equations (E - J), we calculate the quantities ω = 75.6 radians per sec, τ = .083 sec, x_m = .33 in, R_o = 1,500 lb, y_m = .234 in, t_o = .0113 sec. Using these values in equations (A) (B) (C), we get

$$x_1 = .33 \sin 75.6\ t,$$

$$\dot{x}_1 = 25 \cos 75.6\ t,$$

$$T_1 = 1000 + 660 \sin 75.6\ t,$$

$$x_2 = .25 + .234 \sin 75.6(t + .0113),$$

$$\dot{x}_2 = 17.7 \cos 75.6(t + .0113),$$

$$T_2 = 1500 + 468 \sin 75.6(t + .0113).$$

Graphs of these curves are shown in Figure 4. Figure 4a shows the displacement x (Branch 1 and 2) of W as a function of t—this is also the B spring deflection. Figure 4b shows the velocity $\dot{x}$ of W. And Figure 4c shows the thrust T on transducer head H—this is also the compression in the B spring.

In the next section, we discuss the results of this design upon the performance of the thrust stand.*

DISCUSSION OF RESULTS

Operation of Thrust Stand with Buffer Springs

The effects of the buffer springs upon the appearance of the T versus t curves may be seen by comparing the alarming I and C peaks shown in Figure 2 with the corresponding peaks marked I and C on Figure 4c. To simplify the present theory of the buffers, damping was neglected (following the usual "High Q" design procedure). Actually, the T versus t curve shown in Figure 4c settles down to the constant value R_o after the transients die out in much the same fashion as the original startling T curve of Figure 2.

To bring out still more clearly the effects on the T curve produced by buffer springs, a dashed curve marked B on Figure 2 is shown which typifies the new T versus t curves to be expected when buffers are used. It will be noted that no excessive or alarming peaks occur and that the gap is completely eliminated. In other words, the buffers civilize the transient performance of the thrust stand.

It is important to note that the steady value R_o to which the T curve settles down is not the actual rocket or jato thrust R. The actual rocket thrust R is to be calculated from the formula

$$R = \frac{K + k}{K}(R_o - T_o). \quad (13)$$

*In the original paper of which this is a condensation, a number of additional points of design importance are discussed: the optimum design, a family of T curves for different t_P values, choice of pre-load T_o, effects of varying ignitor brisance, guiding factors in choice of buffer spring stiffnesses, effects of damping, effects of shock waves inside the bottle, and others.

The complete article, made of more complete expository nature at the request of the Power Plant Laboratory, WADC, contains many details not included in the present condensed version. This paper may be obtained from: Mr. Emil Malick, Manager, Rocket Fuels Division, Phillips Petroleum Company, McGregor, Texas.

In this formula, K is the stiffness of the B buffer, k is the stiffness of the S buffer, T_o is the initial thrust on the pressure transducer head H, and R_o is the steady value of T. T_o is large in the present case and must be measured just before the rocket is fired (it might well be checked afterward also). R_o is also to be determined experimentally as the steady value of T on the T versus t curve. These remarks are made because uncritical operators of the thrust stand might (and sometimes do) assume that R_o is the actual steady rocket thrust. It is not.

* * *

LABORATORY METHOD FOR SIMULATING SHOCK TO MAGNETIC GRADIOMETERS

R. C. Lowry, U. S. Navy Mine Defense Laboratory

While evaluating a very sensitive instrument as a possible mine location tool, it was noted that the equipment under consideration frequently was out of adjustment when it arrived aboard ship. In order to determine the cause of this misalignment, a series of tests was designed and completed to check the equipment when exposed to a variety of light shocks. The general statistical methods and procedures followed in gathering and processing data are outlined in this paper.

The Underwater Ordnance Locator Mark 2, which was developed by the Naval Ordnance Laboratory during World War II as a means for the recovery of lost experimental torpedoes, has been considered as a means for location of enemy mines, and a system for its effective use was to be developed. This equipment was considered to be free from serious defects, because during the intervening years since its development few complaints had come to light concerning its performance. When the system requirements were determined, it was found that the mine location problem was more stringent than first supposed, and the detector would have to consistently operate at peak performance to be successful. Spurious excitation would have to be kept to a level of about four microgauss per foot, and the detector would have to be able to maintain this low background for weeks at a time.

The Mark 2 Detector is a magnetic gradiometer which consists roughly of two magnetometers which are fixed at opposite ends of a tube which serves to fix a spatial separation, house and support the units, and provide a means for making fine adjustments to the units. The magnetometer outputs are compared electronically, amplified, and are ultimately presented to the user by a dc meter for interpretation. The extreme sensitivity of the instrument makes the interrelationships of all components, either mechanical or electrical, extremely precise.

The detector proper is contained in an aluminum tube which is six-ft long and about four-in in diameter. It has a cable port at one end, and weighs about 45 lb. Figure 1 is a photograph of a detector being handled by a man of about average size. After making all adjustments to the required precision, and transporting the detectors to the point of use, it almost invariably was found that the adjustments had been affected, and the detectors could not attain the required low levels of spurious excitation. No idea or theory could satisfactorily explain what had happened between the alignment area and the ship, since extreme care was always exercised in handling. In beginning the investigation of the cause of this problem, a process of testimonial solicitation was initiated.

The results of this effort were far more amazing than satisfying. One person told of a freak accident in which a detector had been catapulted a height (estimated) of thirty ft in the air, falling to the beach. He reinstalled the unit without checking for possible damage, and found that it worked as well as before. Some other testimony was to the effect that the detectors

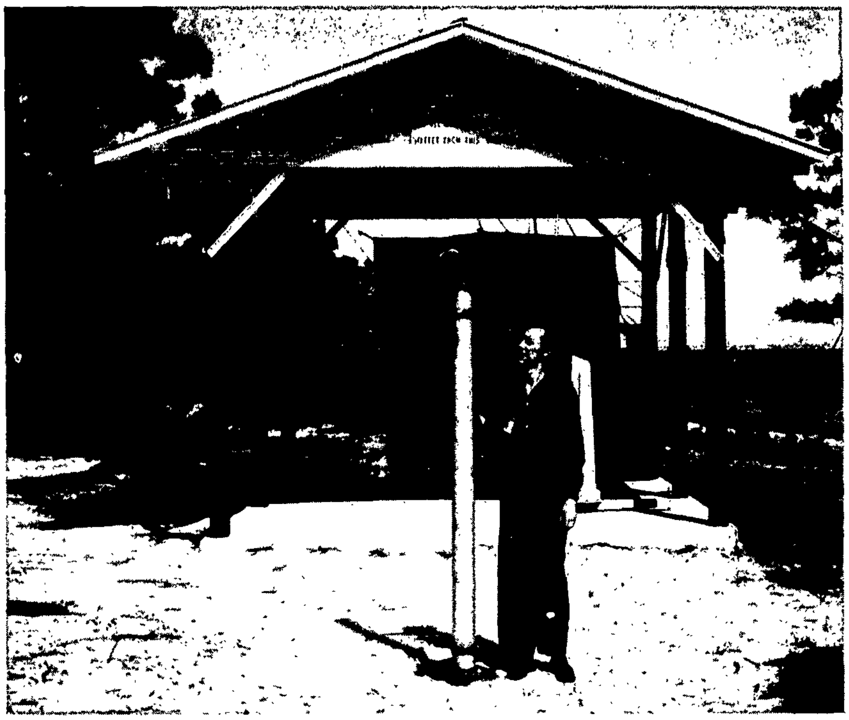


Figure 1 - Mark 2 detector

were not shock sensitive at all, but that temperature and moisture were almost entirely to blame, and told of experiences which led to acceptance of that idea. After two weeks of listening to such reports, it was difficult to hold any firm convictions concerning anything at all related to Mark 2 Detectors. The lack of precise and unbiased observation was quite evident, and the need for some specific tests was clear.

In preparation for the tests, a number of factors had to be considered carefully, both from the purely technical and from the purely economic viewpoint. The complexity of the equipment and the difficulty of its adjustment had been a major factor in determining the seriousness of the problem. The time required to make adjustments was costly, and in addition, special facilities free from stray magnetic fields had to be used. The tests would have to be conducted out of doors at some distance from the laboratory. From a purely technical standpoint, the following factors were to be investigated:

1. Magnitude of impact,
2. Direction of travel of the impact,
3. Duration of the impact,
4. Comparison of the results of the above factors for different detectors.

It was assumed that the experience which led to the belief in the shock sensitivity hypothesis was more valid than that which resulted in the climatic disturbance theory. Had this assumption been wrong, the test should so indicate by failing to show significance for any of the above factors.

Some other limitations had to be imposed for economic reasons. The tests must not result in damage to the detectors, because of the high cost and scarcity of them. The peak acceleration imposed and the duration of the impulse had to be within the range which was expected to be encountered operationally. In testing for effects or impact direction, it was necessary to choose a sufficient number of directions so that peculiar characteristics could be missed. Finally, since the time required would be considerable, the test procedure should be simple enough for a subprofessional assistant to carry out with limited supervision.

Preliminary tests were conducted to determine the best approach to the problem. The simplest manner for imposing shock appeared

to be to raise the detector a calibrated distance and drop it. The control of direction is easily managed by dropping the detector in various orientations. The control of deceleration time is easily handled by choice of the surface upon which the detector is dropped. From the preliminary tests the following ranges of the variables were selected:

(1) The impacts would be obtained by drops between 2-1/2 and 6 ft into a flat bed of sand; the 6-ft drop resulted in a measured acceleration of 15 g with an impulse length of about 0.2 sec.

(2) The orientations would consist of six radial positions plus two axial ones, the six radial positions separated by 60 degrees rotation about the longitudinal axis.

No drops were to be attempted with the longitudinal axis varying from the vertical or horizontal positions because the accelerations measured where one end of the detector struck the earth before the other were found to be always less than when both ends landed simultaneously.

It was decided to make all drops into dry sand and assume that the deceleration time would be essentially constant for all degrees of impact. The size and shape of the detector made this assumption valid when the detector was dropped in any of the six horizontal orientations, and it was decided to ignore the difference when the vertical drops were made. Actually, when dropped in dry sand of considerable depth, the impact in vertical drops would be less and the deceleration time longer due to greater penetration of the bed. Where the sand bed is shallow and backed up by a firm surface, the impact is made far more severe.

Once the above decisions were reached, the next step was to consult a statistician. He studied the test requirements, and submitted a design for the tests. The statistical experiment design called for equal gradations in the two variables (1) impact and (2) orientation. Since eight orientations had been selected, the range of free-fall distances was divided into eight segments varying in half-foot steps. It was recognized that the individual magnetometers would yield separate misalignments, so that each drop or test would yield two separate observations. Figure 2 is a grid of 64 squares upon which the possible combinations of the two variables are entered and numbered. The degrees of impact and the various orientations are given code symbols for shorthand purposes

1. $I_1 0_1$	2. $I_1 0_2$	3. $I_1 0_3$	4. $I_1 0_4$	5. $I_1 0_5$	6. $I_1 0_6$	7. $I_1 0_7$	8. $I_1 0_8$
9. $I_2 0_1$	10. $I_2 0_2$	11. $I_2 0_3$	12. $I_2 0_4$	13. $I_2 0_5$	14. $I_2 0_6$	15. $I_2 0_7$	16. $I_2 0_8$
17. $I_3 0_1$	18. $I_3 0_2$	19. $I_3 0_3$	20. $I_3 0_4$	21. $I_3 0_5$	22. $I_3 0_6$	23. $I_3 0_7$	24. $I_3 0_8$
25. $I_4 0_1$	26. $I_4 0_2$	27. $I_4 0_3$	28. $I_4 0_4$	29. $I_4 0_5$	30. $I_4 0_6$	31. $I_4 0_7$	32. $I_4 0_8$
33. $I_5 0_1$	34. $I_5 0_2$	35. $I_5 0_3$	36. $I_5 0_4$	37. $I_5 0_5$	38. $I_5 0_6$	39. $I_5 0_7$	40. $I_5 0_8$
41. $I_6 0_1$	42. $I_6 0_2$	43. $I_6 0_3$	44. $I_6 0_4$	45. $I_6 0_5$	46. $I_6 0_6$	47. $I_6 0_7$	48. $I_6 0_8$
49. $I_7 0_1$	50. $I_7 0_2$	51. $I_7 0_3$	52. $I_7 0_4$	53. $I_7 0_5$	54. $I_7 0_6$	55. $I_7 0_7$	56. $I_7 0_8$
57. $I_8 0_1$	58. $I_8 0_2$	59. $I_8 0_3$	60. $I_8 0_4$	61. $I_8 0_5$	62. $I_8 0_6$	63. $I_8 0_7$	64. $I_8 0_8$

Figure 2 - Grid showing the combinations of orientation and impact

such as I_1, I_2 --- I_n and 0_1, 0_2 --- 0_n. The symbol $I_n 0_n$ does not indicate the product of I and 0, but merely the combination of the two values of the variables in the single test.

To eliminate as much bias as possible from the tests, the order of testing was randomized, Such uncontrolled factors as climatic changes, instrument calibration drifts, and especially the psychological effects of the tests themselves upon the operator, can be influential in the outcome of the tests unless their effects are confused by randomization. The actual process is simple, and can be accomplished by taking the grid of Figure 2 and cutting out all the blocks; place them in a hat, shake well, and draw them out one at a time to obtain the order of testing.

As the complete data comes in, it is unscrambled and a chart such as Table 1 is obtained. This chart shows the actual measured misalignments obtained during the tests for one magnetometer of one detector. Four detectors were tested, so that in all eight charts similar to Table 1 were obtained. The rows and columns are added and averaged. The statistical handling of data is best omitted here because it is laborious and principally arithmetic. It is sufficient to say that from the data of Table 1, the data in Table 2 are calculated. From the data of Table 2 the variables can be tested. The two last columns headed F and F(.05) are compared as follows:

TABLE 1
Misalignments Measured during Tests for One Magnetometer of One Detector

Impacts	Orientations									
	1	2	3	4	5	6	7	8	Total	Mean
1	0.0	1.0	0.5	0.5	0.0	11.0	0.5	8.0	21.5	2.68
2	3.0	2.0	1.0	1.0	3.0	0.5	17.0	18.0	45.5	5.69
3	2.0	0.0	1.0	1.5	0.5	0.0	1.0	0.5	6.5	0.81
4	7.0	1.0	3.0	0.0	0.0	3.5	11.0	11.0	36.5	4.56
5	0.5	0.5	0.5	1.5	0.5	2.5	2.0	4.5	12.5	1.56
6	1.0	2.0	2.0	0.0	2.0	4.0	5.0	0.5	16.5	2.06
7	1.0	0.0	3.0	2.0	2.0	0.0	13.0	2.0	23.0	2.88
8	0.0	2.0	0.0	2.0	16.0	11.0	5.0	4.0	40.0	5.0
Total	14.5	8.5	11.0	8.5	24.0	32.5	54.5	48.5		
Mean	1.81	1.06	1.38	1.06	3.0	4.06	6.81	6.06		

TABLE 2
Calculated Values for Making "F" Test

Source	Sum of Squares	Degrees of Freedom	Mean Square	F Calculated	F(.05) from Table
Total	1213.44				
Impacts	170.50	7	24.36	1.59	2.17
Orientations	291.25	7	41.61	2.71	2.17
Error	751.69	49	15.34		

The F value is calculated from the other data in the chart. The F(.05) value is obtained from a published table of F values. If the calculated F value is equal to the value of F(.05), then it can be stated that the circumstances under which it was obtained, or the variation of that parameter, will produce this effect only once in twenty trials due purely to chance. Conversely, nineteen times out of twenty, the variation of the parameter must cause the fluctuation in the data.

Now observe the application of the test to the data of Table 2. The F value calculated for impacts is less than the F(.05) value. Therefore it can be stated that the fluctuation in the data is due to variation in impact less than nineteen out of twenty times. Should we desire to know just what probability is associated with this result, we might refer to a table of lower F values. Because it is obvious that impact must produce some effect even before testing was considered, it was not desired to consider this effect further. The purpose here is to test for probability at a specific level.

Now consider the comparison of the F value for orientations. It is apparent that the calculated F value exceeds the F(.05) value, so it can be stated that the effect of variation of orientation is quite significant at the (.05) level. That is, the fluctuations in the observed data would be due to variations in orientations more than nineteen times out of twenty, and due to chance less than once in twenty times. By comparison with a table of higher F values, it was established that the calculated value of F was nearer to the F(.01) or 99 percent probability value than to the F(.05) or 95 percent probability value. Thus one may be confident that orientation is a very significant factor in causing detector misalignment for this magnetometer.

One does not cease to analyze at this point, for there are still questions to ask. A similar process leads to verification that the offending orientations are the two in which the shocks were applied along the longitudinal axis, and that all others fail to show a significance. This result was obtained uniformly for all eight of the sets of data, and so proves to be a general

weakness for all detectors tested. On the other hand, the detectors varied in the degree of misalignment produced, with two of the four being quite superior to the other two.

It proved to be no coincidence, for subsequent investigation showed that the two pairs were produced under separate contracts at different times. Also, the oldest, first produced detectors were the superior ones. The study which followed the revelation that axial shocks caused the most damage led to the discovery that the principal weakness lay in the magnetometer construction, and not in the adjusting mechanism which had nearly always been suspected.

* * *

PART IV

INSTRUMENTATION AND CALIBRATION

A TAPE RECORDING ACCELEROMETER FOR LABORATORY PLAYBACK, ANALYSIS AND SIMULATION

J. Upham and A. Dranetz, Gulton Mfg. Corp.

A miniature recording accelerometer has been developed which can be mounted directly on a test structure and has already been used to monitor simulated water-entry impact of torpedoes. A playback device reduces pulse coded information to analogue form for further recording, analysis, or control.

INTRODUCTION

In the past few years tape recording techniques have been receiving considerable use in the measurement of shock and vibration phenomena. In most cases these are multichannel devices designed for general purpose. Further, they must generally be protected from the vibrational environments in order to prevent background effects.

Recently a miniature recording accelerometer has been developed. A compact and integrated system capable of direct mounting on the vibrating structure, this transducer can monitor laboratory simulation equipment as well as provide recoverable environmental tapes for controlling electrodynamic shakers.

The system is already being used to monitor simulated water-entry impacts of torpedoes. It is also being used in some drop tower testing. It can be used in rocket sled studies; in simulation programs of package testing, it offers a new avenue of approach.

Producing a 40-sec magnetic tape recording of acceleration, this system, when provided with a single-loop playback, can also be used to excite electrodynamic vibrators and other simulation devices which can be electrically excited.

The basic system consists of two major components: a tape recording transducer and a playback device. The transducer is a miniature recorder packaged in a housing 4-1/2-in diameter and 3-in high. Containing seismic element, transistorized circuitry, tape transport, and batteries, this unit is fastened firmly to the test structure. It is activated mechanically or electrically, runs at 15 in/sec, and automatically stops at the end of the tape. The unit records accelerations up to ±60 g and has a frequency response of 0-500 cps. Of novel design, this device features pulse-time modulation to eliminate most of the uncontrolled variables in the tape recording system.

The playback mechanism consists of a conventional tape transport mechanism and a demodulator. The purpose of the latter is to reduce the pulse code information to analogue form for further recording, analysis, or control.

RECORDING ACCELEROMETER

Electronics

In the accelerometer itself the general considerations of rapid starting, small size, lightweight, and low power consumption virtually

dictated the use of transistors. Yet the lack of reproducibility and stability of commercially available transistors obviated the use of a conventional approach to a tape recorder design.

The system ultimately adopted is a combination of phase modulation and pulse time recording techniques. It is shown in the block diagram of Figure 1.

reference pulse and the following information pulse.

It should be observed that signal amplitudes are not critical in a system of this type. For all the information is presented in a time code. Since, as will be seen later, the primary information is in the ratio of the time of delay between reference and information pulses to the

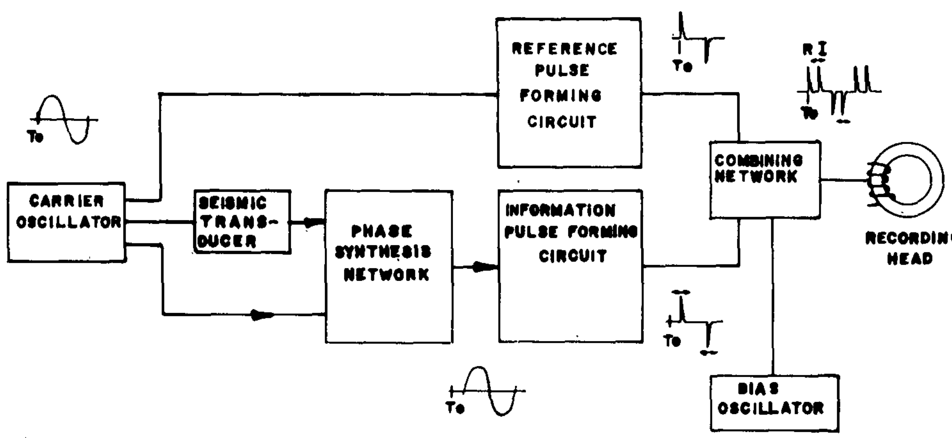


Figure 1 - Block diagram of pulse time recording system

A 1-kc signal generator provides a carrier to an amplitude modulating transducer as well as to a phase reference and a phase synthesis network. The transducer output is also fed to the phase synthesis network. This latter network (which will be described in further detail later) is so designed that the phase of its output with regard to that of the reference signal is a direct measure of the acceleration applied to the seismic system. At zero acceleration this phase angle is 90°. Full-scale accelerations produce deviations of ±20° from this angle.

The two signals are fed to pulse forming networks which consist of over-driven amplifiers and differentiating circuits. These signals are combined in an adding network and applied to a tape recording head. A bias oscillator is used to reduce the signal level required of these pulse networks.

The recorded information consists of a set of 4 pulses occurring at a rate of 1,000 sets per second. The first positive and first negative pulses are reference pulses. The second positive and second negative pulses are information pulses. The amplitude of acceleration is measured by the time occurring between the time between consecutive reference pulses, variations in oscillator frequency or tape speed do not materially affect the stored information.

The actual circuit used is shown in Figure 2. Relatively straight-forward in design, this unit does have several interesting features. In order to provide several volts to the input of the transducer circuit, it is fed from the oscillator tank circuit. Initially transformer coupling was utilized in the amplifiers. Later this was changed to RC coupled networks to reduce size and to reduce phase shifts in the amplifiers. The entire circuit draws 16 ma at 22 volts and is operated from a standard BC photoflash battery.

Transducer and Phase Synthesis

Necessary to the operation of this system is the seismic element - phase synthesis device. In one particular recorder, a differential transformer was utilized.

Ordinarily, a differential transformer normally contains a single-excited primary winding

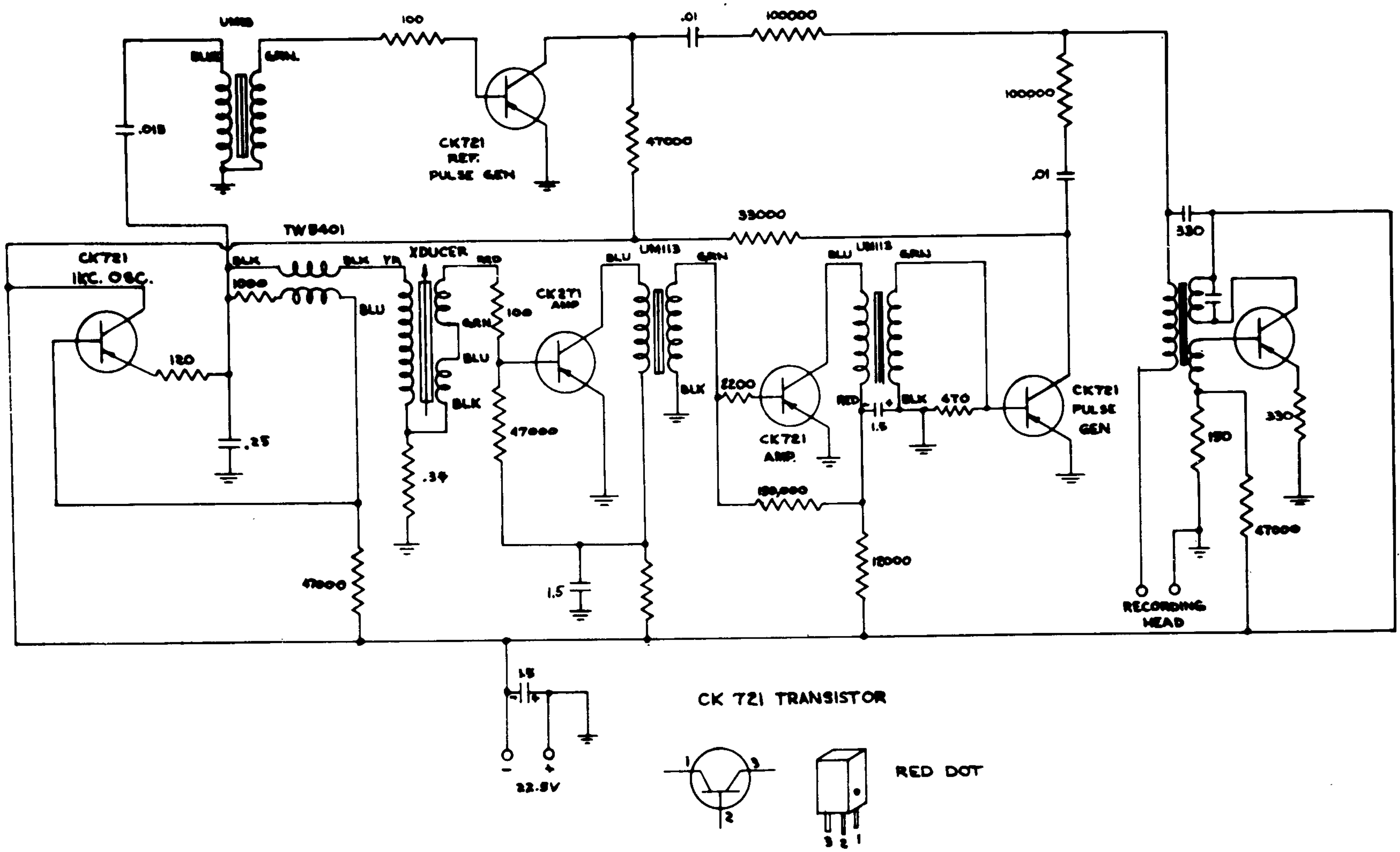


Figure 2 - Recording unit - self recording accelerometer. All resistors in ohms, UM113-Stancor, all condensors in MFD unless otherwise noted. (*Value depends on sensitivity - manganin wire approximately 1-1/2 in long.)

and two secondary windings, all wound concentrically on a tubular form. A magnetic core is positioned in the center of the coil form, and its position controls in linear relationship the relative outputs of the secondaries. When this core is coupled to a high-frequency seismic system, the magnitude of output of the transformer becomes a direct indication of the amplitude of acceleration, and the phase with respect to that of the primary current becomes a direct indication of the sense of the acceleration.

As shown in Figure 3, the primary winding of the transformer is a part of the oscillator tank circuit. The voltage e_1 appearing across the two opposing connected secondaries is combined vectorially with a portion of the primary reference signal e_R. The resulting vector e_i makes a positive or negative phase angle θ with e_R which is approximately proportional to the displacement of the core. A second reference voltage e_r which is approximately 90° ahead of e_R is also shown. This is adjusted to exactly 90° by a phasing network which compensates for the finite Q of the tank circuit.

calibrating resistor such that the ±18° limit is not exceeded.

A simple arrangement is also possible for use with resistance wire accelerometers. This is shown in Figure 4. Here the bridge output is a current passing through the primary of transformer T_2. This output passes through a 180° phase reversal at zero acceleration. Thus the current is either in phase with the voltage across R_c or 180° out of phase with this voltage. Since the voltage produced by T_2 leads the primary current by 90°, the output voltage, which is the sum of the two quadrature voltages, will vary in phase in accordance with the bridge balance condition. Thus the phase of the output will be a direct measure of the acceleration being measured.

Figure 5 shows a partially exploded view of the entire recording accelerometer. At the far right is the cover plate, upon which is mounted the transistorized network along with the seismic transducer. The latter is a fluid damped unit encased in a cylinder 1-in diameter by 1-1/4-in

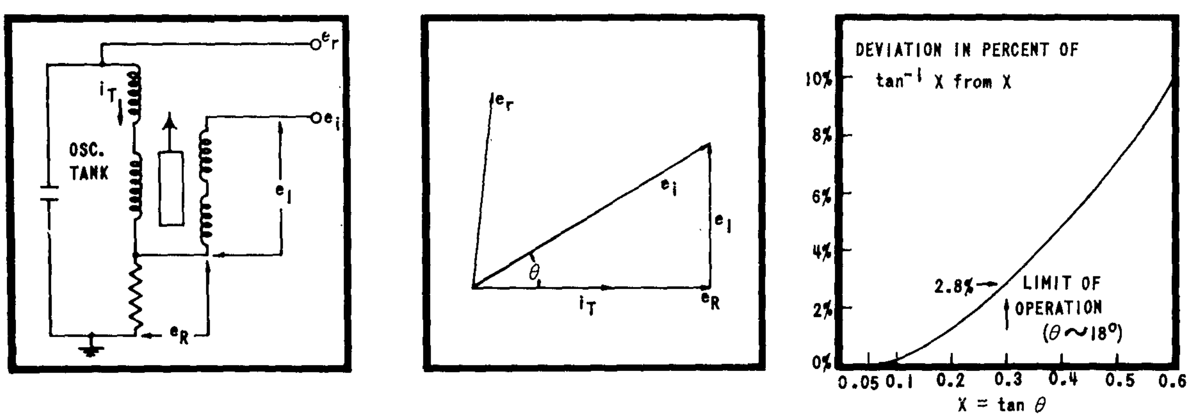


Figure 3 - Operation of phase modulation network

Although the voltage e_1 is proportional to the core displacement, it will be noted that the phase angle θ is determined by the arc tangent of e_1, so it may be considered as proportional only through that angular range where the tangent and the angle are nearly equal. The curve at the right of Figure 3 shows the percentage of deviation from that equality as the phase angle is increased. The practical limit is chosen commensurate with an allowable error of 2.6 percent. This is shown to be a phase angle of ±18°. The full-scale range of the device is controlled by a choice of the

high. This fits on to the top of the open cylinder, shown in the center.

Tape Transport Mechanism

The tape transport mechanism is an electrically driven device operating from 5 penlight cells. No mechanical governors are used due to the accelerations encountered. Instead, the motor characteristics are such that at accelerations of 60 g, the increased friction of the mechanical system is compensated by an

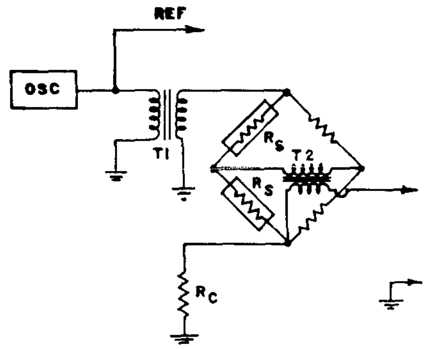


Figure 4 - Phase sensitive transducer utilizing strain gauge

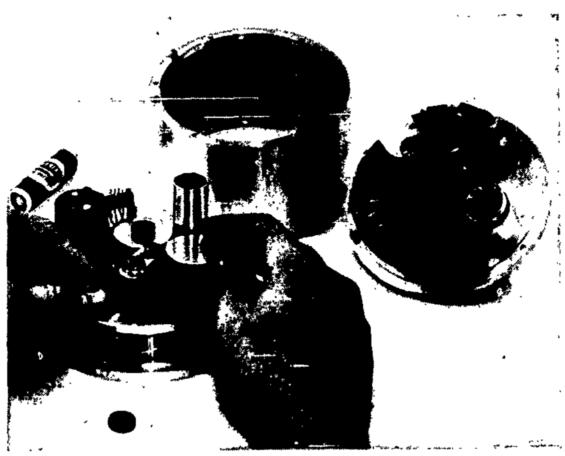


Figure 5 - Recording accelerometer

increase in output torque. Under these conditions speed variations do not exceed 10 percent. Several other unique features are used in this device. Spool friction is obtained by the use of split radial bushings to minimize effects of axial accelerations. The rubber idler is held against the capstan by means of a fixed displacement rather than a fixed force, and a pressure pad is used to hold the tape against the recording head.

PLAYBACK

The playback device consists of two components, a conventional tape transport mechanism and a demodulator. A picture of the combined unit is shown in Figure 6. For feeding electrodynamic shakers the playback is replaced by a single loop playback transport mechanism. The major purpose of the demodulator is to reduce the pulse time code to analogue form.

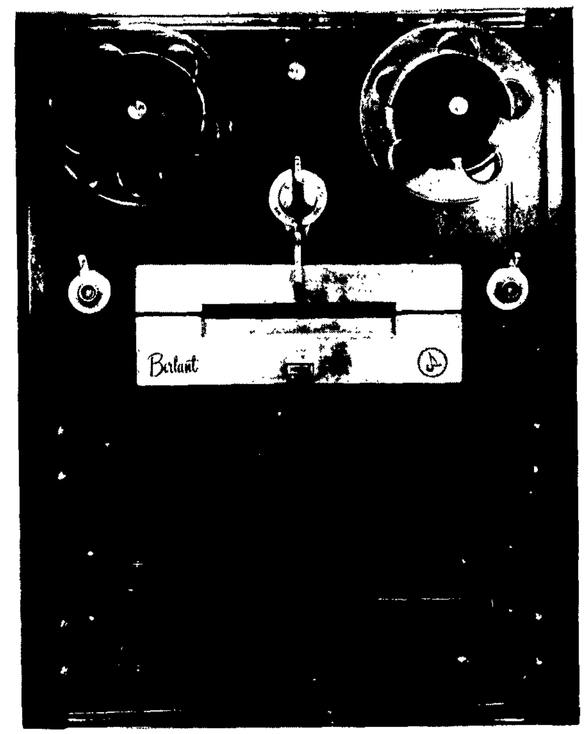


Figure 6 - Playback system

Operation of the demodulator can be best described by following the block diagram of Figure 7.

The output of the playback head is fed to an Eccles-Jordan circuit which produces a square wave timed to the reference pulse. At the same time the signal is also fed to a second bistable generator which is controlled by a blanking circuit. This latter circuit prevents triggering from the first (or reference pulse), but allows triggering from the information pulse.

The two square waves, both of equal and constant amplitude, are fed to a ring modulator. This produces an asymmetrical square wave of 2-kc fundamental frequency, the degree of asymmetry being a function of the delay time. After filtering, the output is a direct measure of the acceleration seen by the recording transducer.

CONCLUSION

The tape recording transducer described above is the result of a 3-year development

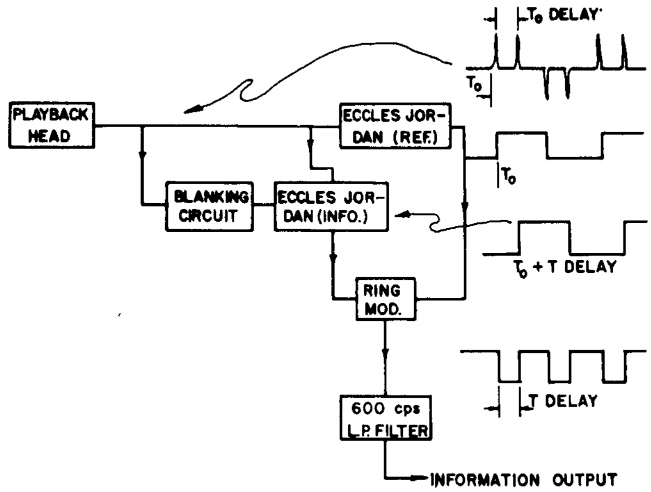


Figure 7 - Block diagram of playback system

cosponsored by the Air Force and Gulton Mfg. Corp. While this system was originally designed for parachute testing, it provides a means for certain laboratory simulation where cables cannot be used. One of the more interesting applications is in package testing. In this instance a three-channel system would, within a slightly increased package, provide a record of accelerations in three orthogonal directions. In addition, the availability of a magnetic tape provides an excellent means of controlling simulation equipment to duplicate the actual environment for which a component must be tested.

* * *

DIRECT READING PROBABILITY METER

D. D. Meyer, McDonnell Aircraft Corp.

This paper describes the development and use of a statistical amplitude probability meter. It is a device which will provide the statistical amplitude probability distribution of a forcing function. By this means randomness is proved or, if the function is not random, will give insight as to the amount of periodicity present.

The present use of nonperiodic vibration in evaluating the quantity level or reliability of electronic components is limited by the lack of instrumentation to evaluate these forcing functions. There is discussed herein a method of determining the statistical amplitude probability distribution of this type of forcing function.

The method of measuring the probability distribution is based on the following consideration:

Let Figure 1 be any arbitrary waveform and the problem be the determination of how much of the total time T, the waveform, spends between two definable limits (y) and (y_1 + dy). An expression for this relationship can be

$$P\,(y)\,dy = \frac{\Sigma\,t}{T}$$

where

P (y) dy ≡ the probability that the waveform spends some time between (y_1) and (y_1 + dy), t ≡ the total increments of time the waveform does spend between (y_1) and (y_1 + dy).

It is apparent from the above equation that the basis of design must be a system that will provide the average of the total time the waveform spends between the limits (y_1) and (y_1 + dy).

The waveform to be analyzed is presented on an oscilloscope using no horizontal deflection.

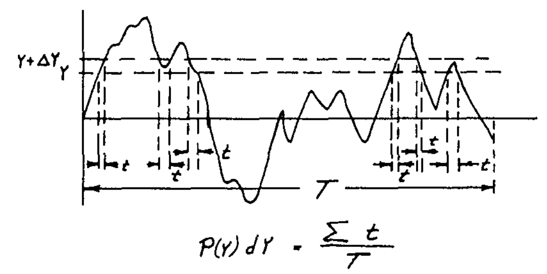


Figure 1 - Random forcing function

This produces a single vertical line, which may vary in light intensity along its length. The average light at any point is directly proportional to the probability of that specific amplitude at that point. A photocell is used to convert light intensity to voltage which can be read from a meter and plotted, or can be recorded in a number of other ways to give a distribution curve directly.

This device is useful in determining the randomness of nonperiodic forcing functions.

PREPARATION OF OSCILLOSCOPE

The faceplate must be removed so the photocell can be mounted directly on the face of the CRT. In order that the sampling interval, dy, be definitely determined, the CRT face must be

masked to present a slot to the photocell of the proper width at right angles to the waveform deflection, Figure 2.

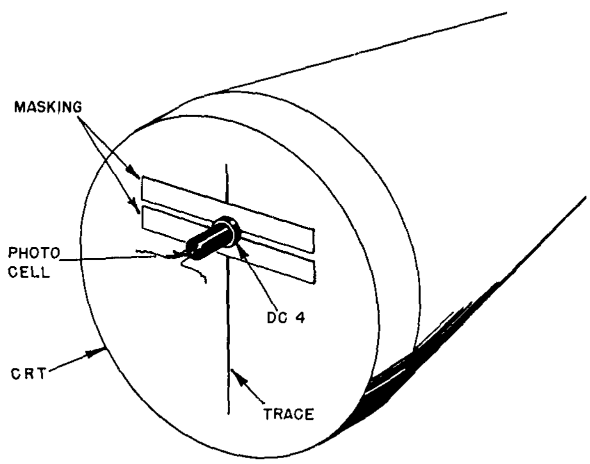


Figure 2 - Preparation of oscilloscope

Slot width was chosen as one-tenth in. This is a practical size to measure and is close to the window size of the photocell used, giving near maximum sensitivity.

Before mounting, Dow Corning No. 4 is placed on the photocell window. When mounted, this grease-like substance fills the volume between the photocell and the CRT face. The "DC 4" acts similar to a "light pipe," improving light transfer by reducing reflection from the cell window.

CIRCUITRY

The circuit is that shown in Figure 3. The photocell is connected in series with a dc supply and a load (R_1). Capacitor C_1 is used to average out small abrupt changes in probability voltage, giving a smoother curve. Resistors R_2 and R_3 provide for mixing a positioning voltage with the waveform to facilitate sampling over its entire magnitude.

PHOTOCELL

Photocells have an inherent characteristic termed fatigue. Fatigue is used to describe the initial drift in photocell output. To minimize the effects of fatigue, the waveform must be moved past the sampling interval very slowly.

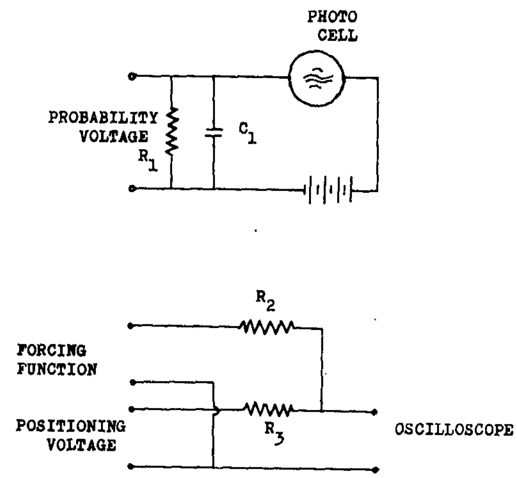


Figure 3 - Circuitry

The cell output is able to follow slow changes in probability very closely. Very abrupt changes are attenuated not only by photocell fatigue, but by the time constant, R_1C_1.

OPERATION

The voltage waveform to be analyzed is presented on the oscilloscope as a vertical line. The light intensity at any given point on the line is determined by the rate at which electrons strike the phosphor. Beam density is constant, determined by the oscilloscope parameters. Therefore the light intensity at a given point is due directly to the time that the beam rests on that point. This is shown in the curve of Figure 4.

The curve of Figure 4 was obtained using a 1,000 cps sine-wave input. The time the spot rested in the sampling interval was changed by varying the sine-wave amplitude. The time spent in the interval, dy, is inversely proportional to the rms voltage. This is true only when the peak sine wave amplitude is many times the sampling interval, and the sampling interval is at or near the zero axis.

With the sampling interval at the zero level, the frequency response of the system can be demonstrated, Figure 4. In this case, the amplitude must be held constant, while the frequency is varied. The upper frequency seems to be limited by the oscilloscope amplifier, while the saturation point of the phosphor causes the voltage to fall off at lower frequencies.

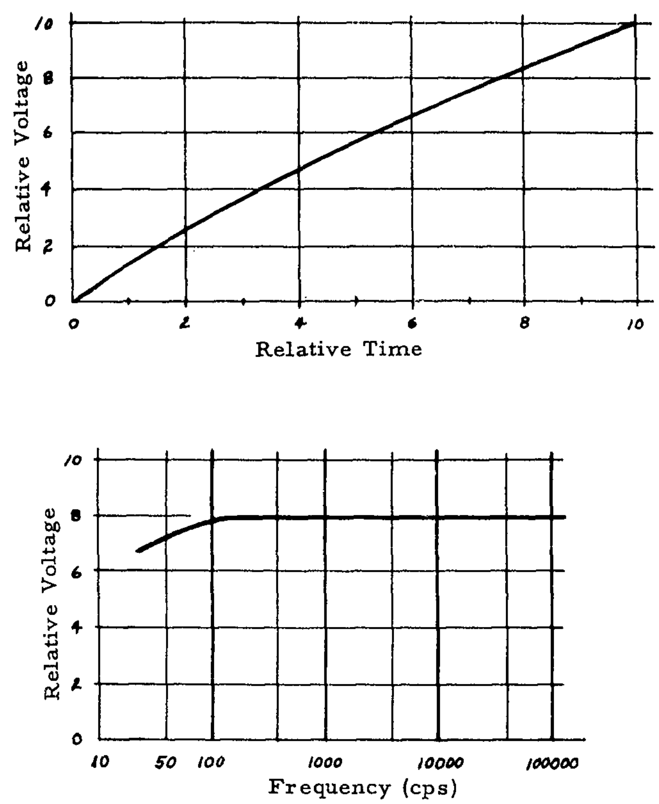


Figure 4 - Response

By positioning the waveform vertically, the photocell can sample the light intensity over the entire magnitude of the waveform. The positioning voltage can be a variable dc for point-by-point meter readings, or a linear waveform (i.e., a saw-tooth or a triangular wave at low frequency) will provide continuous data for plotting. A saw-tooth waveform with a period of 100 sec was used.

There are a number of plotting methods that can be used to advantage. The probability information can be plotted directly with the Sanborn recorder. (This method is least satisfactory because of the compression of the magnitude coordinate due to any nonlinearity of the saw-tooth or triangular positioning voltage.) A plotting board can be used by applying the probability voltage to one axis and the positioning voltage to the other. (The plotting board curve has the advantage of being independent of positioning voltage linearity.) A dual beam oscilloscope, if used, will provide both analyzer beam and plotting means. For permanent record, the oscilloscope display can be photographed.

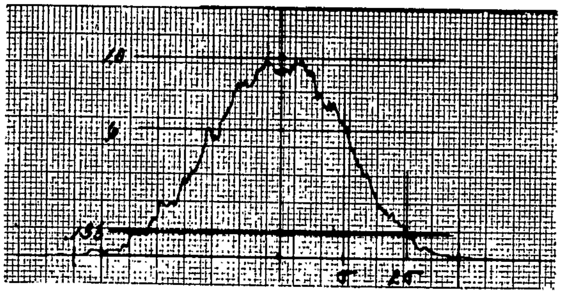
CALIBRATION

As previously stated, the sampling interval determined by the masking must be very small in relation to the peak waveform deflection. A ratio of 1 to 50 was found to be most satisfactory. The oscilloscope masking limits the sampling interval to 0.1 in, therefore adjusting peak-to-peak deflection to 1 in will give a ratio of 1 to 5 peak. By changing oscilloscope step attenuator by 10, the required 1 to 50 ratio can be realized. With the oscilloscope controls fixed, the voltage axis on the recording device can now be calibrated. This is done by introducing a sine-wave input, the nearly vertical portions of the trace indicate the peak voltage.

The probability axis can be normalized by adjusting the oscilloscope beam intensity and the gain of the recorder for a deflection of one at maximum amplitude.

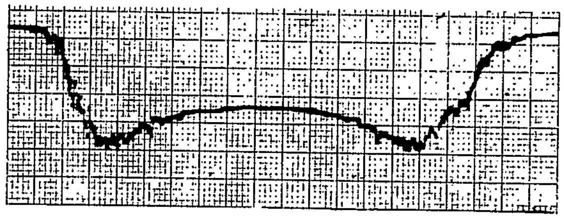
RESULTS

Figure 5(a) illustrates how the resulting curves can be evaluated. This graph shows the



σ	P
.5	.883
1.0	.607
1.5	.325
2.0	.135
2.5	.044

(a)



(b)

Figure 5 - Probability curves; (a) gas-tube noise, (b) noise and sine wave

statistical amplitude probability of the output of a gas-tube noise generator limited in frequency by a 1,000 cycle cutoff band-pass filter. The rms value or standard deviation for this curve is at a point 0.6 of the maximum probability, in this case 2.2 volts.

To prove randomness it is necessary to show that this curve is normal. This is done by the "Ordinate Method of Fitting the Normal Curve." There are tables available listing normal curve ordinates for various values of σ; a partial table has been worked up and is shown here along with Figure 5(a). From this table it can be seen that at twice the standard deviation (σ), the probability should be 0.135 maximum, at one-half σ 0.883 maximum. If these points coincide the curve is normal and the amplitude of the forcing function is random.

Figure 5(b) illustrates the results of a combination of sine wave and noise. The flattened top or double hump is characteristic of this type of function.

DISCUSSION

Grimm, Douglas Aircraft: I would like to ask the speaker whether the meter would allow him to measure the frequency of these vibration noises? In other words, what percentage of the time do we have a given amplitude at what frequency?

Meyer: It is insensitive to frequency anytime.

Voice: I believe that this design is of a given band pass and therefore cannot be assigned to a specific frequency. Am I correct in that?

Meyer: The band width was limited to between 15 and 1,000 cycles.

Wimpey, McDonnell Aircraft: I would like to point out that the same method can be used for finite band widths adjacent all the way through the spectrum. Then you can sample and get the other signatures.

Meyer: Yes, that is true.

Wimpey: I would also like to point out that the probability distribution of any random

distribution process is time-invariant and that frequency, though associated with this distribution, doesn't enter in. If you were to reduce this band width of 15-1,000 cycles you would end up with a single frequency—the probability distribution for a sine wave is the same irrespective of its frequency.

Nitchie, ORL, Penn State Univ: Do you find any particular phosphor or screen type better from the point of view of intensity response?

Meyer: I only used one type. The standard short-persistence blue scope.

Donfor, Diamond Ordnance Fuze Laboratory: My question is somewhat along the same line. I was going to ask about the persistence. If it is fairly short persistence would you run into any trouble say on an especially square pattern where you intend to stay a very large proportion of the time between two levels? Suppose you had something which is essentially a square wave when you dwell most of the time in one spot. Would this bother you persistence-wise?

Meyer: No, it wouldn't.

Wimpey: May I point out that there are cases of distributions that change rapidly in one interval and are constant over another. The sine wave distribution is one example. You can change the nature of your sweep in order to allow more dwell time in the area of rapid changes of probability. For instance, a sine sweep could be used.

* * *

SMALL VIBRATION PICKUPS

Earle Jones, Seymour Edelman, and
Ernest R. Smith, Diamond Ordnance Fuze Laboratories

If equipment is to respond to a simulated environment as it does to the real environment, vibration pickups used to measure the response must not load the equipment significantly. This paper describes a number of small pickups designed to meet this requirement in a number of different applications.

One of the important questions in laboratory simulation is the response of the equipment being tested to the simulated environment. In many cases, it is sufficient to find evidence of gross damage, or to determine the behavior of the equipment before and after exposure to the simulated environment, or to determine the behavior during the exposure. However, a knowledge of the modes of vibration of the equipment being tested and the relative amount of excitation of each mode frequently is useful in itself or as a means for explaining the effect of the simulated environment on the equipment.

If the equipment is to respond to the simulated environment as it does to the real environment, vibration pickups used to measure the response of the equipment to the vibrational portion of the simulated environment must not load the equipment significantly. The pickups shown in the Figure 1 were designed to meet this requirement in a number of different applications. The two large pickups in the background are of the types described in our earlier paper.* The smaller pickup on the left side of the back row is a smaller model of the sandwich type.

The small pickup on the right in the front row was developed to determine the vibrations of a plastic sheet, an application which seemed to us to demand the utmost in low mass and elimination of superfluous parts. It consists of a barium titanate disk cemented to a brass inertial mass by means of conducting epoxy resin. The pickup is attached to the vibrating surface by means of a thin layer of thermoplastic resin. Electrical contact is made to the high-potential side by means of a fine wire soldered to the inertial mass. Contact is made to the ground potential side by means of conducting silver paint which just touches the edge of a thin layer of conducting epoxy resin between the barium titanate disk and the layer of thermoplastic resin which bonds the pickup to the vibrating surface.

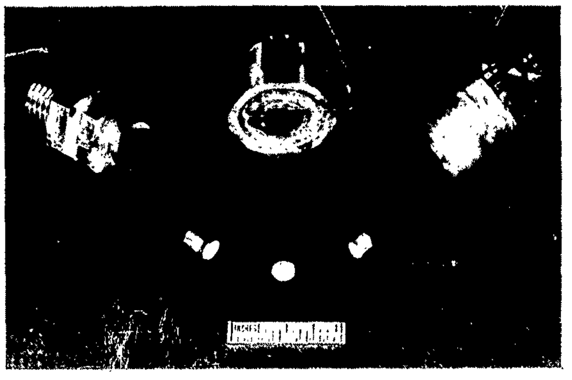


Figure 1 - Types of vibration pickups

Notwithstanding the simplicity of the instrument, a great deal of care in construction and

*Edelman, S., Jones, E., and Smith, E. R., Jour. Acoust. Soc. of Am. 27:728, July 1955

application is required to ensure proper behavior. All mating surfaces must be lapped almost to optical flatness since any irregularity introduces spurious resonance effects. All layers of bonding resins must be pressed as thin as possible. This increases the strength of the bond and ensures that the compliance of the bonding layers is less than that of the barium titanate. The lowest resonant frequency is then set by the inertial mass and the compliance of the barium titanate. The resin material which is squeezed out must be wiped off completely to prevent electrical and mechanical irregularities. The conducting silver paint must be applied carefully under a strong magnifying glass to ensure good dc contact without short circuiting the ceramic.

These pickups are 3/16-in diameter, weigh 0.6 gram, have an output of about 1.5 millivolts per gram, and a capacitance of about 100 picofarads. The first resonance is above 90 kc. Transverse response is about one-tenth the axial response. The thermoplastic resin we have been using gives a good solid bond at temperatures below 100° F. and is soft enough to allow removal of the pickup at about 120° F. The manufacturers, Rubber and Asbestos Corp., have indicated that these temperatures can be increased, if desirable, but so far our applications have been at or below room temperature.

The pickup described above was developed for an application where there was no need to fear electrical pickup or crosstalk. If better shielding is needed, a barium titanate sandwich is used instead of a single disk. The construction can be seen in the pickup on the left on the front row. The sandwich is cemented by conducting epoxy resin. The interface between the two disks is the high-potential side and contact is made to it by means of a fine wire which just touches the edge of the epoxy and which can be seen in the figure. The edge of the sandwich is painted with insulating paint which is then covered with conducting paint so that the outer sides are well shielded.

If a sufficient number of pickups are attached to the equipment being tested, it is possible to monitor the response of the equipment to the simulated environment and to follow the changes in response as the simulated environment is changed.

DISCUSSION

Christensen, Firestone: Would this very simple appearing type of accelerometer be something that was an actual throw away item; stick it on almost anywhere and throw away at each test?

Edelman: Yes, they were developed for just that kind of application.

Christensen: But what about the need for extreme care in handling?

Edelman: I only meant care in manufacture. This means that you have to have a good man to make them, but once you have a good man, he can turn out quite a number.

Siegelman, American Machine and Foundry Co.: I was wondering what control is possible now in circular eccentricity as far as accelerometers are concerned. This seems to have some bearing on the axial transfer sensitivity.

Edelman: As for the pieces for the barium titanate pickups we buy as close as we can to what we think we want, and then we do some shaping ourselves.

* * *

VIBRATION CALIBRATION TECHNIQUES

Mark S. Christensen, Firestone Tire and Rubber Co.

Vibration generators, sensors as well as calibration techniques employed in the laboratory are reviewed and the virtues and vices of commonly used calibration techniques are discussed. A system of optical calibration which has been developed is described.

INTRODUCTION

The purpose of this report is to present a summary of numerous techniques in use for vibration calibration. It does not presume to introduce any revolutionary innovations, but rather, is intended to give a brief discussion of the many systems that have been investigated by the Vibration Section of the Firestone Environmental Test Laboratory. Particularly, the virtues and the faults of each system are listed as objectively as possible. (Numerous reports are available in the literature which go into detailed descriptions of many of these systems; therefore, the comments herein will be relatively brief.) The ultimate selection of any given system will, of course, depend on the likes and the specific needs of the user, but it is hoped that this summary may be of some aid in making the optimum selection.

Vibration testing is coming of age, in most industries, as an inspection means and also as a method of evaluation in developmental design work. The desired vibration may be generated by pneumatic, hydraulic, mechanical, or electrodynamic systems; but regardless of the method chosen, sensors of various sorts must be employed to monitor the vibration amplitudes applied.

A number of systems for measuring vibration have been conceived and perfected, to a greater or lesser extent, over the past years, but all depend on their ability, in one way or another, to sense displacement. (This displacement may be relative, between the various parts of the sensor mechanism, or it may be absolute, related to a fixed point on earth.) The sensors may be classified according to their output characteristic as displacement, velocity, or acceleration indicators. The characteristic selected depends upon a number of factors, particularly the magnitudes of displacement anticipated, the range of frequencies to be investigated, the size and mass of the pickup proper, and the portability and ease of handling of the complete sensor system.

A further classification may be made, according to the method by which the sensor develops its signal:

1. Externally-excited units (variable resistance, variable capacitance, variable reluctance transducers).
2. Self-generating units (piezoelectric crystals, electromagnetic generators).

These systems may be manufactured as portable units for temporary usage or they may be constructed within the framework of a vibration-generating machine. (In this latter instance, the most common type employed is the electromagnetic sensor which generates a voltage signal proportional to velocity.) Dependent upon the test requirements, it may or may not be necessary to concern oneself with absolute calibration.

If investigative or developmental testing is to be performed on a variety of prototype

components, it may be adequate to compare or standardize a group of pickups which are to be employed so that they all read alike. Since developmental improvement is usually a relative matter, absolute values of vibration may be of little importance. However, when subjecting manufactured components to acceptance specification requirements, it is mandatory that the vibration amplitude be calibrated in absolute terms. For this purpose, a simple, practical method must be established for absolute calibration of measurement systems at routine intervals or whenever any discrepancies are suspected.

Included herewith is a vibration nomograph (Figure 1) to aid in understanding the various quantities to be measured. The nomograph is certainly not a new device, but this chart, as presented, is a broad expansion of the region in which most vibration tests are conducted. For those readers who are not intimately familiar with vibration parameters, it is suggested that they make frequent reference to the chart. Note particularly that all scales are logarithmic. In moving to the right (increasing frequency, maintaining constant velocity), acceleration levels are increasing and displacement values are diminishing—at an exponential rate.

Calibration at low frequencies (with large displacements) is an easy matter, even by simple measurements, performed by the naked eye. However, in order to explore higher frequencies, reference to the chart will immediately reveal that extremely small amplitudes are involved, specifically the amplitude at 300 cycles and 10 g being only 0.002-in. It is readily apparent that specialized systems must be devised in order to measure precisely dynamic displacements of such small magnitude. It is possible to employ various accessory sensors (variable-reluctance or variable-capacitance pickups attached to fixed mounts), but it becomes unwieldy to attempt to calibrate an already complex electronic device with another even more complex.

The most basic and direct system for dynamic physical measurement is to resort to optical means, employing an optical instrument to magnify the small areas to be studied and a measurement system which is capable of indicating extremely small displacements. Up to a limit of about 300 cycles, it appears practical to perform physical measurements aided by optical magnification. However, in order to perform calibrations at much higher frequencies, say about 1,000 cycles, displacements are measured in terms of micro-inches. It then becomes essential to perform measurement by means of other systems such as the interferometer. The operation of the interferometer is based upon a count of fringe shifts across a glass plate, illuminated by a monochromatic light source. The system is not very practical for large displacements because of the extremely large number of fringes which have to be counted.

The discussion to follow will deal only with the subject of calibration by optical means and will be divided into four major sections: Optical Instruments, Measuring Systems, Targets, and Illumination.

OPTICAL INSTRUMENTS

Telescope vs. Microscope

The optical unit to be employed may be either a microscope or a telescope, depending mainly on the surrounding circumstances and to some degree on the preferences of the operators. The magnification of the unit should be at least 40 to 50X, but powers much above this may prove of no additional value unless there is an unusually fine optical system which will suffer no losses in definition and resolution at higher magnification powers. (High magnification appears to present more manufacturing problems in telescopes than in microscopes—when additional limiting requirements such as size, maneuverability and working distance are imposed.) Many investigators have employed the microscope for measurement, for the reasons of presumably simpler mounting requirements and fewer problems in the mount regarding its vibration. We have tried both systems, finding no significant advantages for the microscope, but in the case of the telescope numerous advantages such as:

1. The telescope is extremely useful not only as a calibration device, but also as a "mass-less" sensor unit for measuring displacements (in any direction) of various portions of complex test subjects.

2. The telescope has a long, widely-variable working distance which allows extreme freedom of action in selecting the site upon which to set up. In fact, it is ideal for setup in the center of a number of shakers, of different sizes and types, not requiring any fitting to the shaker itself.

3. The telescope allows the operator to sit completely at ease, remote from regions

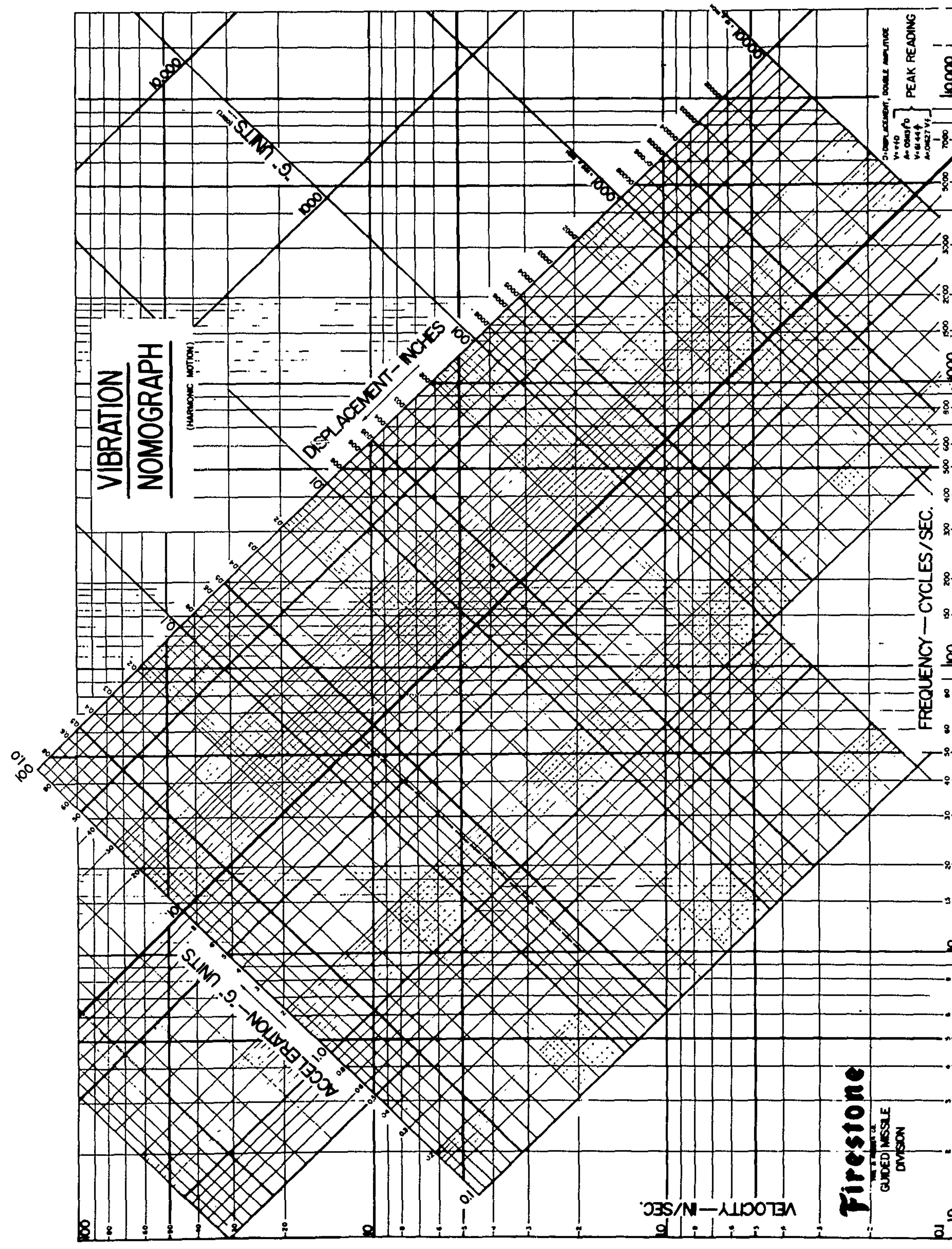

VIBRATION
NOMOGRAPH
(HARMONIC MOTION)
DISPLACEMENT - INCHES
ACCELERATION - "G" UNITS
"G" UNITS
VELOCITY — IN/SEC.
FREQUENCY — CYCLES/SEC.
PEAK READING
Firestone
GUIDED MISSILE
DIVISION

Figure 1

of highest vibration and noise level. This fact is of considerable importance since operator fatigue rapidly becomes one of the major sources of error in calibration work.

4. The telescope allows for simultaneous manipulation by one operator of a remote vibration control console and also the measuring system, thus eliminating any need for "calling signals" between two operators.

5. The telescope, although affected by surrounding vibrations, is capable of measuring same and also is able to measure motions in the presumably "stationary" portions of the shaker body. The advantage ordinarily claimed for the microscope is that ambient vibrations from the surrounding area are eliminated from its mount, since it is usually attached directly to the shaker body. Often the shaker is mounted upon a large concrete block, supported by a spring system of a very low natural frequency (particularly in the case of small shakers which are intended exclusively for calibration work.) It is quite true that external vibrations may be virtually eliminated from the microscope in such a system, but vibrations from the shaker itself may be induced directly into the microscope, with little knowledge of their presence, phase relationship, or magnitude.

MEASURING SYSTEMS

In the optical system selected (most of the following remarks would apply equally well to the microscope or telescope), some means must be incorporated for the actual measurement of displacement. We have tried a number of systems, and the relative merits of each were exploited.

Eyepiece Reticle

The simplest device is a graduated reticle installed in the eyepiece, divided in arbitrary decimal units. Since the magnification of the optical system (and thus the resulting calibration) is dependent upon the working distance of the optical system, the true calibration at the required working distance must be established by accessory means. This calibration may be performed by placing an engraved scale at the target site, or by observing the size of a Johannsen gage block. Rather than replace the target with a calibration accessory, it is simpler to mount a dial micrometer over the vibration table and by displacing the table a specified static distance (as indicated by the micrometer), the number of units of excursion across the reticle may be observed. After calibration of the eyepiece is established, a desired dynamic displacement of the vibration table may be set by matching its total excursion to the required number of divisions in the reticle. The advantages of this system are:

1. Extreme simplicity—no moving parts in the optical system.

2. Ability to measure the top and bottom of excursion simultaneously.

3. Ease of maintaining a constant displacement over a frequency scan by holding the excursion set to a specified number of divisions.

The points cited are distinct advantages, which make this system highly desirable. Accompanying disadvantages, sufficient to recommend further elaboration of the system are:

1. Difficulty in setting any displacement that is not equivalent to a whole number of divisions. This requirement occurs when attempting to calibrate accelerometers at constant acceleration over a frequency scan.

2. Difficulty in measuring extremely small displacements—less than one whole division of the scale. The alternative here is to make extremely small divisions on the scale, but this entails more problems in fabrication of the reticle. Also, it tends to obscure the clear field of view through the telescope, and complicates the study because of the requirement to fix one's attention upon a maze of extremely fine lines.

3. Difficulty of precisely "mating" or "matching" the target to a given line in the reticle.

Eyepiece Micrometer

The eyepiece micrometer is a measuring device which can be installed in virtually any telescope—or microscope. It consists of a

precisely guided carriage which transports a set of cross hairs across the field of view of the eyepiece. Travel of the carriage is controlled by a precision lead screw. Affixed to one end of this screw is a drum, graduated in decimal units about the periphery. A revolution counter is also geared to the screw so that measurements across the full field of travel can easily be read.

In the eyepiece are two stationary cross hairs, intersecting at 90°, to indicate the center of the field and to aid in alignment upon the target. The micrometer carriage contains additional cross hairs which may be mounted in various configurations, but the "bifilar" arrangement is most ideal for calibration work. This arrangement consists of two very close-spaced, parallel cross hairs, whose position in the field of view is determined by the setting upon the micrometer lead screw. The usual manufacturer's spacing between the bifilars is approximately 60 microns (0.0024-in).

The bifilar micrometer is extremely versatile, in that it may be used with any of the targets to be described later. In measuring vibration, the micrometer is set to read one limit of excursion and then the other; the difference between readings is the measure of total amplitude. The great advantage of the bifilar over a single travelling cross hair is that it eliminates the need of setting to the "upper" or "lower" edge of a point or line target. To make a reading, the micrometer is set so that the target appears equally spaced between the filars. A proper setting of the bifilar, for several types of targets, is given in Figure 2.

The fact that the target is positioned between filars eliminates the necessity of determining its static "size", with the inherent errors incurred in subtracting this size from all subsequent measurements. The bifilar is readily adaptable to all types of illumination. It is immeasurably superior to a single filar in accomplishing settings under adverse circumstances, such as: dim light, excessive contrasts in target brightness, and "fuzziness" of targets.

Optical Micrometer

The optical micrometer is a very interesting device composed principally of a precisely ground, flat, plane-parallel disc of glass which can be attached to the fore-end of any telescope for the purpose of measuring. The disc is mounted in a ring which contains bearings across its centerline, for supporting the disc and allowing it to be rotated about this line (Figure 3). If the disc is rotated in either direction through some angle away from perpendicular (to the optical axis), refractions occur at both air-glass surfaces and the observed image is displaced from the optical axis, but the new line of sight is always parallel to the optical axis. A calibrated dial is used to rotate the disc, the calibration being dependent only upon the thickness and refractive index of the glass selected. There is mounted in the eyepiece of the telescope

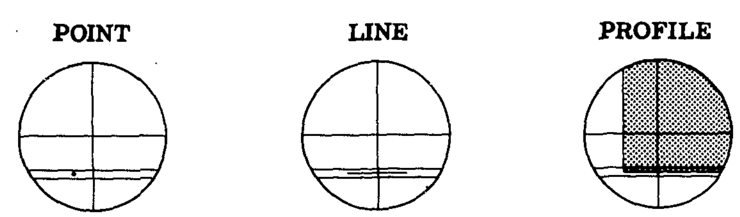


Figure 2 - Bifilar settings

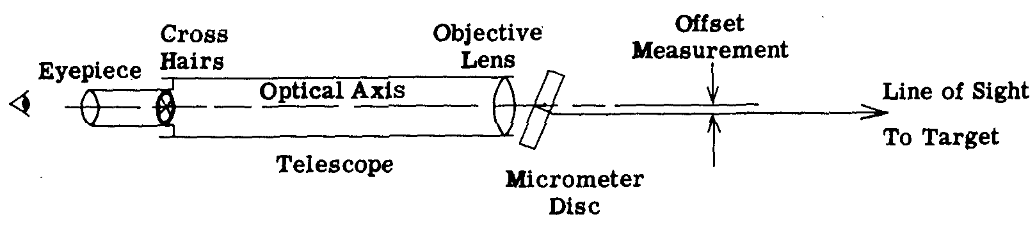


Figure 3 - Optical micrometer

a set of fixed cross hairs defining the optical axis of the tube. In vibration calibration, the micrometer is rotated to "read" first one limit of excursion, and then the other. In the case of both limits the target image is deflected from its true position and brought into coincidence with the cross-hair reference on the optical axis. The difference between readings is the measure of displacement. The most outstanding characteristic of this device is its manner of dealing with parallel lines of sight, so that calibration of measurement is independent of magnification power or working distance of the telescope.

Through the courtesy of a local optical supply house, we borrowed an optical micrometer in order to explore fully its characteristics and learn its suitability to vibration calibration work. The unit was mounted on a surveyors level of approximately 42X magnification, working at a distance of about 8 ft. Our initial usage of this instrument was exactly in the manner described above. A most interesting deviation from normal usage was achieved by mounting the unit horizontally off-set toward one side of the telescope so that the micrometer disc covered only a part (about one-half) of the objective lens. In this case, two separate images are seen, one being the direct image through the telescope only, the other being the image through the micrometer disc, linearly displaced by whatever setting is on the calibrated dial. (Ideally, for this application, the micrometer disc should be cut in half, so as to cover exactly half the objective lens. However, for experiment only, the shape of the disc, or the percentage of objective covered, is of surprisingly little importance in achieving the effect described.)

Illustrated are three pictures seen through the telescope, set up in the manner just described (Figure 4).

In both cases, the picture actually consists of two separate images, exactly superimposed. The third picture is obtained by moving the micrometer to displace the images, so that the top of one image (either one) is exactly "matched" to the base of the other. The setting upon the calibrated dial is now the true reading of total excursion. This operation does not require exact positioning of the target to any particular portion of the field of view (as with a reticle) and does not require separate readings at the top and bottom of excursion.

The advantages anticipated for this system were realized in some instances, but several disappointments and definite surprises were also incurred. In the final reckoning, no matter what system of measurement is to be incorporated with the telescope, it must be realized that vibrations of some sort are bound to be present in a floor-mounted telescope, even though it be remotely located from the shaker to be observed. These vibrations may be generated by the shaker under study, by other shakers in the vicinity, or by machinery or vehicles in the plant.

It had been anticipated that with the optical micrometer system the effect of these vibrations would be nullified, because of the fact that the measurement is made by matching two images (even though the "reference" may also be subject to vibration), rather than by matching one image to a presumably "stationary" reference cross hair. However, our tests revealed that with the micrometer mounted on the telescope, and with the whole subjected to vibrations through the mount, the mode of the micrometer unit was different from that of the telescope. The result was that the "height" of the image (each image) was not constant—it would wax and wane at some arbitrary frequency which appeared to be a beat-frequency between the various parts of the optical system and the

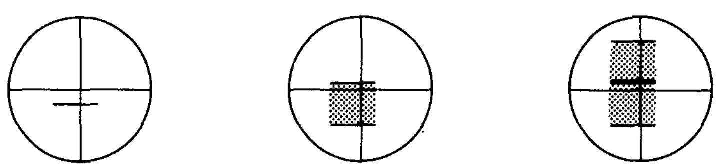


Figure 4 - Micrometer settings

The first picture shows the target line at rest, with the micrometer adjustment set at zero. The second picture shows the target in motion, the micrometer setting still at zero.

frequency of the exciting force. This phenomenon rendered impossible any attempt to match the edges of two images (or even the edges of one image to a cross hair). It should

be pointed out that the manner of mounting the micrometer was not adequate—that by enhancing the structure it would likely be possible to eliminate such a phenomenon. This may be true, of course; but, regardless, other pertinent circumstances also detract from the system.

We had visualized that it might be easier, and more exacting, to match the edges of two images rather than to match either edge to a cross hair. This proved quite untrue because of the rather "fuzzy" appearance of the edges of the image. It is extremely difficult to determine when two fuzzy edges are brought into exact contact (neither overlapped, nor failing to touch). This feat actually proves more difficult than matching a fuzzy edge to a reference line in the eyepiece.

Another problem is the handling of the micrometer dial. Since the micrometer is mounted at the front of the telescope, the operator has to reach an appreciable distance to manipulate the dial, while attempting to maintain a very delicate touch. As mentioned previously, operator fatigue over several hours of calibration work is a critical factor and everything possible should be done for his comfort—and consequent ability to perform accurate work.

To summarize, the advantages of the optical micrometer (if they can be fully realized) are:

1. Total excursion of the target can be read by a single setting of the micrometer dial.

2. Two images of the target are compared for measurement of displacement; it is not necessary to match one image to a presumably stationary reference cross hair.

3. Calibration of the measuring system is established by the manufacturer. It is independent of the working distance or the magnification power of the telescope used.

4. Exact positioning of the target is not required. So long as it is clearly visible, the target may be at practically any place in the field of view.

The disadvantages of this system are:

1. The optical micrometer must be attached by an extremely rigid mount to preclude any differential motion between the various parts of the system.

2. The position of the micrometer at the front of the telescope adds tedium and fatigue to its usage.

3. It is exceedingly difficult to match precisely the fuzzy edges of two target images to each other, or to a reference cross hair.

TARGETS

Assuming that the optical and measuring systems have already been selected, the next problem is the fabrication of a suitable "target" to be studied. The characteristics of the target must be duly considered not only from their own isolated aspect, but also for their compatibility with the rest of the system. The general requirements are that the target be exquisitely small, sharp, and "clean" to ensure precise readings of very small displacements, and that it be simple in form and "easy on the eyes." The final selection, of course, is also affected by personal preference of the operator, as no one particular target can truly be judged "the best".

Although "illumination" will be discussed in detail later, it is necessary at this point to divide the targets into two classifications: those which will be viewed by reflected light and those viewed by transmitted light.

Emery Grit

A well-known, easily available target is emery grit, in the form of abrasive cloth. A small patch of cloth may be glued to the vibration table and illuminated by a spotlight mounted at a slight angle off to one side. The viewing instrument is set up approximately perpendicular to the surface of the cloth and sharp focus is fixed upon the emery. The picture obtained is a dull, dark field, speckled at intervals by numerous brilliant points of light. The random orientation of the multitude of emery crystals on the cloth is such that relatively few of them are properly aligned to reflect light to the observer. From the numerous pin points of light presented to view, it is not particularly difficult to select and fix attention upon one bright spot of a size and shape that appeal to the eye, properly located in the field of view to be within the scope of the measuring system. The strong feature of the emery target is the fact that it is easily available; however, several disadvantages are also encountered as follows:

1. The random distribution of emery crystals on the cloth is such that on repetitive set-ups of the viewing instrument it is most unlikely that any one "target" will ever be duplicated. This decidedly does not aid uniformity, reliability, and repeatability of measurement.

2. The "size" of the grit which is chosen as the target is of definite significance in any measurements to be made. The measurement system must be employed to determine this size (with the inherent errors of this determination) and the amount must be deducted from the total value of any displacement measured, in order to determine the true displacement. The smaller the displacement being measured, the greater will be the percentage error introduced by the static size of the grit. Of course, it is possible to employ finer and finer grades of emery in order to minimize this error, but as the density of bright spots in the field of view is correspondingly increased, it ultimately becomes impossible to concentrate on one spot with sufficient acuity to use it as a target.

There is no doubt as to the extreme value of devoting considerable study to emery grit targets, but the limitations are such as to advocate development of a more refined target system.

Machined Profile

A machined profile, either on the vibration table itself or otherwise specially fabricated, may be employed as a target. Various sharp profiles naturally exist on the table and also on components which would likely be under test. However, the difficulty is that profiles which would ordinarily be available are extremely rough, as seen under high magnification, and this roughness is a hindrance to precision measurement. One simple target of this sort is a razor blade, either as viewed from the flat face or upon edge. The sharpness of the ground edge presents a fine profile, but extreme care must be taken that no resonance of the blade itself occurs at any frequencies to be encountered during calibration.

Probably the most perfect profile target would be a Johannsen gage block affixed to the table by appropriate clamping means. However, if special precautions are not observed, the difference in light intensity from a bright reflecting surface (the profile) to the remainder of the field (which is usually dark) is extremely hard on the eyes. Consequently, it is difficult to compare a reference line in the instrument to a sharp profile as the intensity of the picture abruptly changes from harsh glare to dark field. For such study, it would be advisable to experiment with various shades of gray or possibly some other soft color in the background upon which the profile may be easily differentiated when compared to the reference line.

Point

A single, small point may be used as a target, either an opaque point on a transparent field or a transparent point on an opaque field. In either case, the problem is to make a point sufficiently small for use under magnification. To apply any sort of opaque point on a transparent field (glass) of sufficiently small size appears well nigh impossible. Regardless, whether or not this would be possible, further considerations (especially glare and eyestrain) indicate that it is not worthwhile. However, a transparent point on a dark field has very strong appeal, principally because of eye-ease and consequent minimum operator fatigue. Again the problem is to make a small point. It is not possible to drill a sufficiently small hole for this purpose by standard factory methods; however, it may be capable of attainment by jeweler's techniques.

Another possibility is to apply an opaque coating over glass and to scratch clear a small point for transmitting light through the coating. It is remarkable how inadequate most such efforts appear under high magnification!

However, one very fortunate happenstance exists in the case of mirror surfaces deposited under high vacuum. Under even the most perfectly controlled conditions, it seems that most such mirrors have numerous microscopic surface flaws, all of which transmit light. It is possible to study these flaws under high magnification and, noting the smallest flaw perceptible, circle it, and solidly black in the remainder of the field. This is the most perfect point target we have been able to achieve. We have not found it to be appropriate for calibration work, but it is an extremely useful target for other purposes, particularly for studying spurious, nonlinear motions of vibration tables.

Scribed Line

A scribed line appears to us to be the most ideal target, but the method of scribing the line

presents many difficulties. For example, a line may be ruled upon a white card with a ruling pen; but it is impossible to produce a line, by this means, of sufficient sharpness for use under high magnification.

A line may be scribed by a sharp cutting tool, particularly by a diamond point onto the surface of a metallic or glass target. Although this appears to be the best system, the edges of the cut are always slightly torn and rough, no matter how fine a line is attempted. Consequently, even the finest of lines is not absolutely "clean". It is necessary, then to accept this fact and learn how best to deal with it.

For targets viewed by means of transmitted light, we have scribed lines, as fine as possible, on opaque, mirrored-glass surfaces. We have found it possible to achieve a much finer and cleaner line in this manner than by attempting to make a dark line on a transparent surface. In addition to the advantages of rendering a sharper line, the light line on the dark field also adds to operator comfort as the total glare in this method is at a minimum.

If a dark field is to be studied, however, accessory means must be provided for illuminating the reference cross hairs in the telescope. (A very simple solution we have employed is merely to mount a white card toward one edge of the field of view close to the telescope. As a result some stray light is cast into the optical instrument.) A line of this sort is most simple to study, and considerable benefit is achieved by use of a line (rather than a point) because the alignment of the measurement system can then easily be oriented to the direction of motion of the target.

ILLUMINATION

A number of illumination systems may be employed with whatever optical and measuring systems are incorporated, each having certain distinct advantages.

Color

The simplest and most obvious illumination is steady-state, white light—either natural or from tungsten filament sources. The major problem encountered in using white light is the spectral dispersion of colors after emanating through the very small slit of the scribed line discussed above. This problem may be avoided by using a monochromatic source, such as an arc-discharge lamp, available in a wide choice of colors. Reference may be made to physics tables listing the wavelengths of different colors, to determine the ultimate limit of measuring ability under light of various colors. It can readily be seen that in this respect the blue end (short wave lengths) would be more desirable than the red end of the spectrum. However, wavelengths toward the blue-violet may be injurious to the eyes of the operator. (Actually, the amplitudes to be measured are far greater than the limitations imposed by wavelength, so the color selection is of little consequence in this regard.) An easy compromise is to pick a color midway in the spectrum—namely, green. Further reference to tables listing color sensitivity of the eye show that visual acuity for a given amount of light energy is greatest for this color. Moreover, it is frequently considered the optimum color for eye ease.

Source

With any of these light sources the next point to be considered is the nature or character of the source of the light. Generally speaking, it is possible to provide diffused, point-source, focused or collimated light from any of the sources mentioned above. We have devoted some study to each type and have concluded that these factors appear to be of little importance and probably can be ignored unless they have some specialized association with the remainder of the test setup employed. However, it is certainly worthwhile to devote some time to experimenting with these factors to determine whether anything might be gained in a particular setup.

Other Characteristics

The mention of arc-discharge lamps will, perhaps, suggest use of stroboscopic light. We have devoted considerable study to stroboscopic illumination and find it extremely worthwhile for certain vibration studies but have not found it particularly useful in calibration work.

Consider a moment the appearance of a line, under steadystate illumination, vibrating at high frequency. It appears to be a broad "band" of light, more intense at each extreme of motion because of the apparent "dwell" (zero velocity) at the extremes. The size of this block of light is quite definite and a measurement can readily be made by comparison to an eyepiece reticle or by measuring the range of motion with any of the measuring systems described above.

However, if stroboscopic light is to be employed, the apparent image of a single line may be made either to stand still or to move slowly at any cyclic rate desired by the operator. The advantage in this case is that a very clear single line (rather than a band of light) can be studied with ease.

If the two extremes of motion are to be measured, however, at apparent standstill or in slow motion, it is exceedingly difficult to ascertain that the absolute extremes of motion are truly being observed. Determination of these two extremes under stroboscopic conditions is dependent upon persistence of vision rather than a constant display of total motion as would pertain if steady-state illumination were employed.

Some very worthwhile applications may be made with stroboscopic light for setting a given displacement, or for maintaining a constant displacement of a set amount while traversing a frequency scan. In this application a fine etched-scale glass target may be mounted upon the vibration table and a displacement equal to a whole number of divisions of the scale may be set by observing under stroboscopic slow motion when two lines representing the desired displacement are brought into repeated coincidence. In this case, it is necessary to employ a variable frequency power source for the stroboscope so that the illumination may be "synchronized" to a frequency near that of the motion to be observed.

If arc-discharge lamps are to be used as the light source, it is evident that a stroboscopic effect will be encountered at the power line frequencies, and multiples thereof, whether desired or not. If this effect proves troublesome, then it will either void the possibility of calibration at such frequencies or otherwise will require that the arc lamp be powered by a variable-frequency source.

It may be worthwhile, also, to mention polarized light. We have examined this but have not devoted much time to it, as it appeared that no particularly desirable effects were achieved through its use. However, a pair of filters—one at the light source and one at the optical instrument—may be used for very effective control of the intensity of the light observed, by shifting the orientation of one filter in reference to the other. Generally, the source intensity of arc-discharge lamps is not easily controllable because of their design for fixed voltage; therefore, it is advisable that some means be incorporated for varying the intensity of illumination presented to the eye. A simpler system for controlling intensity is to install an iris in the light beam close to the source.

CONCLUSION

Following is a description of the vibration calibration system as we have evolved it for our laboratory. (Figures 5, 6 and 7.) For the optical instrument, we have a telescope of approximately 50X magnification power. This incorporates rack-and-pinion focusing of the tube containing the rear elements and a sliding extension tube for the front elements so that a wide variation of 3 to 150 ft in working distance may be achieved. The versatility afforded by such a wide range of working distances is invaluable to us.

The measuring system is a bifilar eyepiece micrometer with fixed cross hairs oriented at

Figure 5 - Over-all view of calibration setup

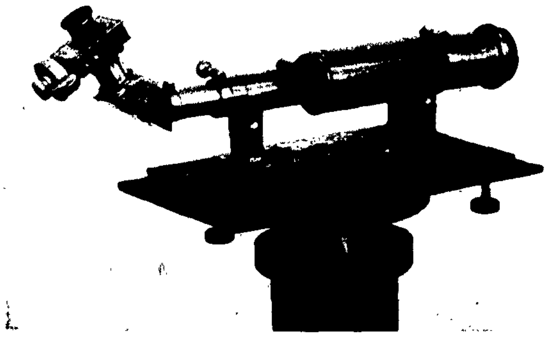


Figure 6 - Measuring telescope

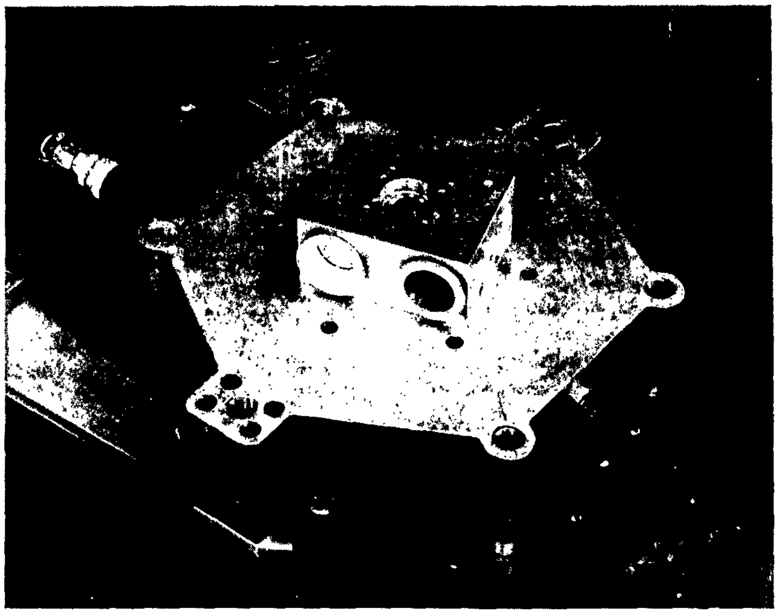


Figure 7 - Target block

90 degrees through the center of the field and a bifilar measuring line which can traverse the entire field. The mount which includes this measurement system can be rotated through 360 degrees, thus allowing horizontal or vertical measurement of displacement, or measurements at the random angles of motion occurring in complex test components. (No matter which measurement device is selected, it is exceedingly important that it be capable of rotation as described. Thus the measuring axis is easily brought into perfect alignment with the axis of motion to be measured.)

Our measurement system is mounted to the tube atop a 60-degree erecting prism to accomodate operator comfort. The tube is set up at a height of 35 in above the floor to correspond to the table level of all vibration exciters in our laboratory.

By use of the 60-degree offset prism, the operator is enabled to sit comfortably in an arm chair with his elbows firmly supported and chin cupped in hands. This position makes for minimum fatigue and is also the easiest and most sensitive manipulation of the micrometer dial which is located directly below his chin. Always stressing the importance of operator comfort, we believe this system is not to be excelled.

The target employed is a glass disc of 27-mm diameter with a silvered surface and a pair of transparent scribed lines which admit light through the target. (The disc is mounted in a hole drilled through a block of aluminum which affords firm support. The top of the aluminum block is drilled and tapped for various types of accelerometers which are to be calibrated.) The disc serves simultaneously as the target and also as an absolute calibration for the measurement system. See Figure 8 for clarification of this point.

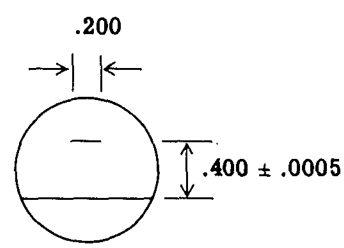


Figure 8 - Target disc

In our procedure, we align the measuring filar with the long line of the target to establish proper orientation of the system. A reading of the distance between lines is made with the vernier micrometer, to establish the calibration of the measuring system at the working distance in effect. Then, according to preference, the operator may devote his attention to either of the two lines for vibration measurements.

The thickness of the line itself is approximately 0.0005-in, this being considered by the maker of the target disc as the minimum which would be reasonably practical for use in this system. We believe it would not be particularly desirable to request a finer line because the target line should occupy a significant space between bifilars to aid in centering.

The light source used is a high-pressure, arc-discharge mercury lamp (rich in green radiations) with three successive green filters to isolate the single, green line at a wavelength of 5460 A. These filters are mounted in a bracket directly affixed to the lamp housing. Following this is a small condensing lens. This condenser is used to concentrate the illumination at the target, but no specific effort is made to bring the light to focus upon the target.

After a lengthy period of usage, we find no difficulties with any portion of the system. As a routine operation, we now perform calibrations on our vibration signal monitoring systems and accelerometers over a frequency range to approximately 300 cycles. In most instances this calibration is performed at a constant acceleration of 10 g so that the displacement has diminished to 0.002-in at 300 cps. (Higher frequencies may be attained by increasing the acceleration at this same displacement limit.)

It is difficult to define precisely the accuracy of our system; however, we can refer to terms of reliability and repeatability. We find that an operator can, over five or more successive trials, repeat a reading with an error not exceeding 0.0005-in for any setting. It can be seen from this that accuracy on relatively large amplitudes is extremely fine; at a displacement of 0.005-in, we believe it is about ±10 percent.

DISCUSSION

J. Osmanski, Cook Electric Co.: What was the smallest amplitude you measured in these tests?

Christensen: As you can see on the vibration nomograph we measured down to 32 thousandths of an inch at 300 cps. At 10 g you hit 2 thousandths of an inch. This is the smallest measurement we feel we can make very well.

Osmanski: What about the degree of accuracy?

Christensen: At the 2-thousandths level we reached a limit of usefulness with the accuracy becoming poor. At 5-thousandths we have an accuracy of about plus or minus 10 percent; at 2-thousandths it is probably around 25 percent. Again I wish to stress that the purpose of our system is not precise calibration to the standards of a precision laboratory, but more or less routine calibration to ensure proper performance of our systems.

Sowell, Hollowman Air Development Center: Did you consider the autocollimator method of measuring this vibration displacement?

Christensen: Yes, it had been considered. We devoted quite a lot of argument to it. We had an optical technician of considerable experience in our group; he proposed the autocollimator many times. However it cost far more money than we wished to invest. I believe it requires a massive foundation and its installation would have involved about four times the cost we actually incurred in establishing our system. It would be undoubtedly very good for more precise calibration, however, we were concerned with simplicity of setup and with a relatively low-cost system; a system which any of the men in our group could operate.

Shuffler, Convair, Ft. Worth: I was wondering if you had considered the Wild precision level?

Christensen: Yes, we did use that. Firestone worked rather closely with the Jet Propulsion Laboratory of Cal Tech and they have a Wild level. I practiced quite a bit with it there, but again this has an optical micrometer at the forward end of the telescope. Also the magnification power of that instrument was a little less than we wanted. It had a magnification of around 40X at the working distance we were using. There was a single filter in the eyepiece. It is a beautiful surveying instrument, but more complex than we wanted in our work.

Shuffler: The graduations down to a thousandth of an inch appealed to us. Do you have any information on how accurate this might be?

Christensen: No. I don't for sure. Some of our group use a K & E Micrometer; they are estimating 40-thousandths and achieving very good accuracy with that system. It seems to be a very beautiful instrument.

Shuffler: One other thing I wanted to ask about. Have you any experience with these devices which take signals either from a pick-up on the table or from the armature of the shaker and use them to provide pulses to a strobe-light?

Christensen: Yes, we use this quite a bit for studying table motions. It's a wonderful device. One is made in California called the Slip-Sync. Maybe you are aware of that one. We pick up the frequency of our shaker itself and put it through the Slip-Sync.

Actually the full frequency of the table minus one cycle per second is applied to the light source, so that the motion we see is a relative slow motion of one cycle per second. Very beautiful it is for studying random motions of points on a vibration table, but we did not find it very good for calibration itself. The main reason why we don't employ stroboscopic light of any sort for calibration is the fact that it's difficult to know when you have the top and bottom of movement. Without a stroboscopic light you see a continuously presented bank of light and you know for sure that you have a top and a bottom. If you are looking at slow motion you are depending now on persistence of vision and you are trying to remember when you saw the top and when you saw the bottom. It's quite disconcerting.

Edelman, National Bureau of Standards: I would like to put in a word for the stroboscopic system. In ours the motion is stopped or moved at the operator's will. You can tell very nicely which is the top and the bottom, because we move it there and stop when we get there.

A question I wanted to ask was how do you know when your telescope axis is exactly at right-angles to the motion of your target both in the plane of the axis and at right angles to the plane of the axis of the telescope? Do you have some way of leveling and aligning?

Christensen: In the eyepiece of the telescope we have a pair of cross hairs mounted at 90 degrees and the vibration target is inserted in the block of aluminum I showed you on the table. Now, the eyepiece can be rotated in the telescope and the disc containing the target can be rotated on the table.

The glass target disc has some additional minute imperfections as well as the two lines. These show as a few random points. Now we set the table into fairly large motion and observe some of the lines drawn by these random points. With them we can establish when we have a alignment truly parallel to the motion of the table. These lines on the target should move not only always parallel to themselves but also at right angles to the lines marked on the disc. If the lines of the disc are slightly inclined the successive motions may be parallel to each other but we shall see the difference in orientation compared to the cross hairs in the eyepiece.

Maybe I have not made this too clear, but it's a matter first of all of establishing the orientation of the target disc to the motion of the table and next aligning the eyepiece exactly to the disc again.

Edelman: How about if you have a motion away from the telescope as well as up and down?

Christensen: You have got me there. It's a very good point and for sure this would get us into lots of troubles at higher frequencies. But at frequencies lower than 300 cps we haven't found those motions to be any significant percentage of the motion we are actually measuring. Also we don't have any difficulties with table resonance at frequencies below 300 cps.

Orlacchio, Gulton Mfg. Corp.: I don't think that most of the people are having too much trouble with the low frequency calibrations. One of the big problems now with vibrators is to be able to get an absolute calibration of these in some manner above 500 cps since most of the military specs are now going to higher and higher frequencies. I was just wondering whether you had considered doing anything at the higher frequencies or have you done anything using, say, the interferometer technique or some form of that type of calibration.

Christensen: No. Over these past two years I have been trying to visit other people who do experiments with the interferometer, to find out how it works, but we have so far no requirement to do this ourselves. The main thing we do is establish on a routine basis the general accuracy of our measuring systems. Most of our work is concerned with frequencies of 500 cps or less. If we have to get up in the higher frequencies at some later date, we also may be driven to the interferometer, but so far we have not had the distinct need and up to now we have avoided it.

Shuffler: Have you experienced any difficulty with your vibration table actually drifting during its oscillation about any particular point?

Does it ever drift above or below that point? Have you been able to see this in your system?

Christensen: This is a good argument for the eyepiece reticle. You measure the top and bottom of the motion simultaneously, whereas if you have the optical micrometer or eyepiece micrometer you measure the top at one moment and sometime later you measure the bottom. It's true that during this time, drift of amplitude or drift of center position could occur, but so far we have had no indication of this. Higher frequencies might lead to some troubles of that sort—as might voltage instability in the power supply of the shakers. We have a very stable voltage supply in our area. Maybe this is in our favor. At any rate, we have never determined we are troubled by this.

* * *

PART V

ELECTRON TUBES

DEVELOPMENT OF A WHITE-NOISE VIBRATION TEST FOR ELECTRON TUBES

John Robbins, Sylvania Electric Products, Inc.

A white-noise vibration test has been developed for the vibration evaluation of electron tubes over a wide range of frequencies. White-noise vibration is explained theoretically and compared with sinusoidal vibration. A practical test method is described, and details are presented on the white-noise generator, vibration test equipment, and on methods of reading the tube noise output.

INTRODUCTION

The increasing interest in the vibrational performance of vacuum tubes, as in guided missiles, has created a demand for test methods capable of evaluating tubes over a broad range of vibrational frequencies at relatively high values of acceleration. One possible approach is the use of sine wave vibration, swept over a wide spectrum such as 200 to 2000 cps. This test is too time consuming for large-scale testing. A rapid sweep rate will produce erroneous noise readings, but if the test is performed correctly it is useful as a design test. Another technique is the use of magnetically recorded multifrequency vibrational conditions especially for specific known installations; but this method lacks generality. The idea of white-noise vibration has often been considered and this approach was followed in a wide-frequency vibration test developed under Bureau of Ships Contract NObs 8023. This test is not restricted to any particular tube application, utilizes simple meter-type noise readings, and is believed to be adaptable to large-scale testing. The test equipment developed is for subminiature tubes, but the test procedure itself is also suitable for any other tube type or component.

EXPLANATION OF WHITE NOISE TEST

The standard fixed frequency, fixed g-level vibration test is defined simply and concisely by the conventional methods. The specified g level (for example 15 g) refers to the maximum value of acceleration experienced during sine-wave vibration at a single specified frequency (for example 40 cps). This type of specification is not directly applicable to a white-noise vibration test which necessarily has peak values of acceleration occurring at highly irregular intervals, while the intermediate acceleration values comprise a random path in time instead of a sine wave. White-noise acceleration is properly a random-noise function and can be represented by procedures utilized by S. O. Rice in his article "Mathematical Analysis of Random Noise" (BSTJ, Vol. XXIII). The approach is as follows: Let $e(t)$ be the noise function. Consider the instantaneous values of $e(t)$ to be plotted versus time. The resulting plot may be analyzed as a Fourier series over an arbitrary time interval of length T, resulting in one set of Fourier coefficients for the following equation.

$$e(t) = C1 \cos(w1t - \phi 1) + C2 \cos(w2t - \phi 2) + \ldots + Cn \cos(wnt - \phi n), \quad (1)$$

or

$$e(t) = \sum_{n=1}^{N} Cn \cos(wnt - \phi n) \qquad \begin{matrix} N \longrightarrow \infty \\ \Delta f \longrightarrow 0. \end{matrix} \quad (2)$$

If the analysis is repeated many times for different intervals of time, all of length T, different values of phase angle, ϕ, result. $\phi 1$, $\phi 2$, $\phi 3$, . . ., ϕn will each assume values which are

randomly distributed between zero and 2π radians. Thus in Eq. 2

ϕn = random phase angle distributed uniformly over the range $(0, 2\pi)$,

$C_n = [2W(f_n)\,\Delta f]^{1/2}$,

$W(f_n)\,\Delta f$ = value of power represented by band width Δf at frequency f_n,

$w_n = 2\pi\, f_n$,

Δf = band width associated with the nth frequency component,

$e(t)$ = random noise function. This function can be considered to be either acceleration or its voltage analog.

Equation 2 shows that a random-noise function may be viewed as containing a wide spectrum of frequencies and is therefore called "white noise" in comparison to white light which contains all visible wavelengths.

The rms of $e(t)$ in Eq. 2 is easily shown to be

$$e_{rms} = \sqrt{\sum_{n=1}^{N} \left[\frac{(C_n)^2}{2}\right]} \qquad (3)$$

where $N \rightarrow \infty$ and $\Delta f \rightarrow 0$.

Equation 3 is equally valid when N is a finite number of components, each represented by its rms value and each associated with a specific band in the spectrum. The rms value for the entire spectrum is then

$$e_{rms} = \sqrt{\sum_{n=1}^{N} (En)^2} \qquad (4)$$

where $En = C_n/\sqrt{2}$ = rms value of nth component and where the sum of the N components equals entire spectrum. If each component contains the same amount of energy as represented by E, the equation becomes

$$e_{rms} = E\sqrt{N}. \qquad (5)$$

Before proceeding further it may be useful to list the items that control the character of a white-noise function. These are

1. Band width and location of the frequency spectrum,

2. Total energy content,

3. Distribution pattern of energy within the spectrum,

4. The degree of clipping of random peaks.

For the test developed at Sylvania, it was considered most useful to have the energy distributed equally per octave, where one octave means a spectrum interval having a ratio of 2:1 between upper and lower frequencies. This energy distribution realistically balances the relative importance of the different constituent frequencies. Now, if the upper and lower frequencies of the total spectrum are f_2 and f_1, the number of octaves (represented by N) may be calculated as follows:

$$2^N = \frac{f_2}{f_1}$$

or

$$N = \log_2 \frac{f_2}{f_1} \qquad (6)$$

or

$$N = 3.32 \log_{10} \frac{f_2}{f_1}$$

Substituting 6 in Eq. 5 yields

$$e_{rms} = E\sqrt{3.32 \log_{10} \frac{f_2}{f_1}} \qquad (7)$$

where E = the rms value for one octave.

Equation 7 can be used to represent acceleration or its voltage analog as obtainable with an ideal accelerometer. This equation can be adapted to any part of the spectrum by choosing values of f_2 and f_1 which define the band area of interest. The use of Eq. 7 determines the total energy content if total band width is specified by f_2 and f_1 and if E, the energy per octave, is specified.

The final item to be defined is the amount of clipping present. In practical applications the random peaks of a white-noise voltage are limited to some maximum value, often determined by a clipping circuit designed for that purpose. These peaks are neither symmetrical nor uniformly spaced, but a peak-to-peak value may be observed on an oscilloscope. The amount of clipping can be specified in several ways, but for the white-noise vibration test developed at Sylvania it is controlled by specifying a peak g value of 15 g.

The other specification parameters used are 100 and 5000 cps for spectrum boundaries, and 2.3 g rms per octave for energy distribution.

This latter value was chosen somewhat arbitrarily but is compatible with the 15 g peak value and is about as high as is obtainable without clipping excessively. The total rms value can be calculated from the expression 2.3 g rms x 5.6, where 5.6 is the number of octaves used. The resultant total value of 5.4 g produces a ratio of peak to rms of very nearly twice the 1.4-ratio that is characteristic of a sine wave.

DESCRIPTION OF EQUIPMENT

A white-noise voltage generator, shaping circuits, and power amplifier are used to obtain a complex voltage input which has its spectrum shaped so that it will cause a vibrator armature to move according to the frequency spectrum required for acceleration. A barium titanate accelerometer mounted on the armature is used in setting up equipment to meet these test conditions. The voltmeter used in metering the acceleration should have a true rms response. The complete system for measuring acceleration (including accelerometer and associated circuits such as a cathode follower and amplifier must have flat frequency response and linear amplitude response within the requirements expected of the acceleration spectrum itself.

The white-noise generator uses a type 6D4 gas tube as the noise source, a type 6AT6 as the voltage amplifier, and a type 12AX7 as a cathode-coupled clipper.

A block diagram of the white-noise vibration set is shown in Figure 1. The noise voltage proceeds from the white-noise generator to a band-pass filter for establishing the upper and lower frequencies, then through a preamplifier for shaping the active spectrum, next through a power amplifier for obtaining the drive needed for the desired magnitude of acceleration, and finally to the armature coil of the electromechanical transducer. This transducer was originally constructed for generating sine-wave vibration at 10 g for 50 - 5000 cps.

The design, similar to one originated by the National Bureau of Standards, has a helical coil type armature moving in a constant magnetic field. The frequency response curve of this equipment must be free of any conspicuous antiresonances. When long lead subminiature tubes are tested, the method of holding the tube disposes the tube leads into two horizontal layers of four leads each, with the leads being clamped at their extreme ends. Attenuation, by lead

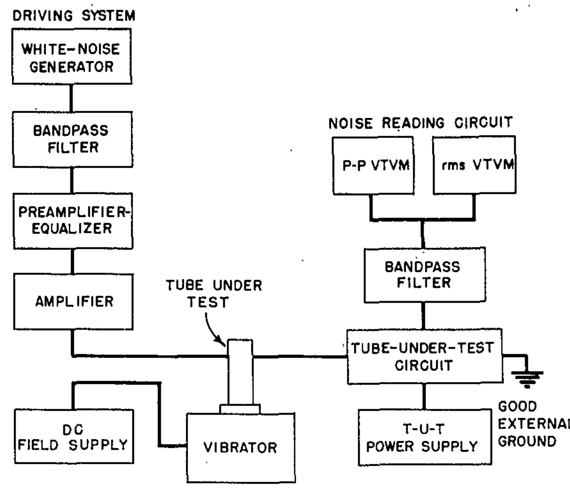


Figure 1 - Block diagram of white-noise vibration equipment (all equipments and cables are shielded and connected together by ground wires)

stiffness, of the high-frequency vibration g values is not a problem because the required amplitude becomes very small (of the order of microinches) for the higher frequencies and because the specification and equipment calibration deal with the acceleration actually experienced by the moving armature and tube. The lower frequencies are more of a problem than the high ones because the angular deflection of the leads at low frequencies becomes significant and thus limits the g level obtainable with clamped leads. Tube holders are currently available for the vibration testing of long-lead tubes with the plane of vibration perpendicular or parallel to the grid major axis.

VIBRATIONAL NOISE OUTPUT OF TUBE UNDER TEST

The vibrational noise output voltage from the tube under test is the complex ac voltage generated across a resistor (usually 10,000 ohms) in the plate circuit in accordance with present specifications for vibration test circuit parameters.

The noise-output voltage does not necessarily have the same band width as the acceleration input, and for this proposed test, noise frequencies above 10,000 cps are excluded by filter.

The different types of readings that were considered for the test were:

1. rms value—A voltmeter with true rms response could be used to obtain a single reading for the total noise over the entire spectrum up to 10,000 cps.
2. Average value—A voltmeter could be used which gives a single reading for the average of the positive values.
3. Sub-band readings—Tube noise spectrum could be divided into sub-bands, each read by method 1 or 2 above.
4. Peak-to-peak value—The noise would have no true peak-to-peak value but a reading could be obtained which is partly a function of the response characteristics of the instrument.
5. Sonic analyzer readings—Limit values could be assigned for the magnitude of noise frequency components as determined by a sonic analyzer. Different limit values may be used for different sections of the band.

Sonic analyzer readings were used for several months in an experimental run of production testing but results were not wholly satisfactory. The test was time consuming because each sweep of the sonic analyzer required one full second, and many sweeps were required to show a good over-all picture of the noise output.

Two types of readings are now being taken and are proposed for specifications. The first is a peak-to-peak value as read on a peak-to-peak V.T.V.M. The second reading is taken on an rms V.T.V.M. which has scales marked in volts rms but is actually more of an averaging device.

It must be remembered that relatively high values of noise can be expected for peak-to-peak readings. For a pure sine wave alone the peak-to-peak value is 2.8 times the rms value and the occurrence of harmonics even in fixed, single-frequency vibration always tends to raise this ratio rather than lower it. Thus a tube with 50-mV rms output for 15 g, 40-cps vibration could read over 300-mV P-P. An even higher ratio of peak-to-peak/rms can be expected for white-noise vibration output. The ratios are sometimes 10 to 1.

The results achieved from this test have been satisfactory and the test is now in use for a line of subminiature tubes currently in small-scale production.

DISCUSSION

A. A. Emmerling, General Electric Co.: I'd be most interested to hear how these tubes are mounted so that you could know for sure what the input was. I imagine you also run destructive tests and I'd like to find out why you developed these tests in terms of power density.

Robbins: They are mounted in an armature by insertion into a hole the diameter of the tube. There is a sleeve that goes around the tube and a screw that clamps down and contracts this sleeve. The sleeve is split, thus providing a tight fit on the tube. The accelerometer is mounted on the moving armature. Incidentally, this is a lightweight vibrator; it is not equipped for vibrating heavy loads.

Emmerling: What kind of destructive levels do you reach with this test in terms of power density?

Robbins: We have not run this test to what we call destruction. For one thing, we were limited in the amplitude that we could obtain. Also we have a variable-frequency fatigue test which we developed as a special destructive test. The white-noise test developed evaluates tubes without being destructive, it only approaches destructive levels.

Voice: I take it, then, that you had no tube failures. I wonder what you get from the testing, itself—if you are always going to have to produce failures to gain knowledge.

Robbins: Maybe I should clarify a point here that might be confusing. The comments on the previous papers presented this afternoon have referred to destructiveness of tests. Most tests performed on tubes are not intended to be destructive but rather to supply information on tube characteristics. Consider heater-cathode leakage. No one expects such a test to be capable of conduction to destruction although tubes might be so tested for experimental purposes by using very high voltages. But one

would still want to have some test, if possible, which can be applied to all tubes to weed out undesirable tubes.

Failures for vibration, of course, depend on the limits set; but I would also say that this test does detect some failures of the type sought in destructive tests. Sometimes we have tested tubes which had failed to show up as failures by any previous testing but did so in this white-noise test. For example, intermittent shorts and sometimes opens would be revealed by the readings. But this test is not designed necessarily to evaluate the tubes to destruction.

Voice, Sperry Gyroscope: I was wondering if you could give us some idea how the military would write a specification, if it ever came to using this white-noise test. What would be the variable that they would ask the various manufacturers to test tubes to?

Robbins: I will tell you what we have set up for our special line of tubes. We specify a maximum g value (15 g), which is similar to the conventional test. And, of course, we specify the frequency covered which I said was 100 to 5000 cycles. And I specified the rms value of energy per octave as 2.3 g rms per octave, thus always defining the acceleration present at the tube.

E. N. Sowell, Holloman AF Base: I have one question. What is the method of shielding the tube magnetically from the electric field of the shaker? I assume that it is an electrically driven shaker.

Robbins: We have the tube shielded by a concentric tubular shield. Also the magnetic field is concentrated well below the tube area.

S. P. Sanders, Bendix Aviation: You used assumed power density levels based on actual cases? Do you have any experimental data or magnetic tape or anything to substantiate this—rather than use a uniform power spectral density?

Robbins: You want to know why we used equal energy per octave instead of equal energy per cycle?

Sanders: That's right.

Robbins: I think I mentioned that we did try equal energy per cycle—or rather equal energy per cycle to the one half power—and the results showed that this was not a desirable test because the higher frequencies dominated the results. That is, all results would be the effects from the higher frequencies and one might as well ignore the lower frequencies altogether. This showed up especially when we used the sonic analyzer in our test to show the frequency distribution of the tube noise output. In order to take advantage of the full width of the spectrum, we wanted to weight it relatively to obtain the best over-all evaluation of the tubes. Also, there is no reason why one should assume the energy should be equal per cycle. It might just as well be equal per octave.

M. Barmat, Glenn L. Martin Co.: I have two questions. The first concerns whether this energy spectrum is related to physical realities, and the second question is, can you give us any idea of the noise expected across the 10,000-ohm load.

Robbins: The first question: We believed the band from 1000 to 2000 cycles, a spread of 1000, should be much more important than a spread from 4000 to 5000, and that more of a logarithmic distribution would be best to show up physical resonances. Does that answer your first question?

Barmat: I was particularly concerned whether any particular aircraft had this spectrum when tested.

Robbins: This test was developed by a tube manufacturer, and as such, we do not have the facilities to conduct actual practical applications ourselves. Also, the purpose of the test was to evaluate tubes for vibration without bringing in specific applications. My understanding is that every application encountered would show up different results. These tests have not yet been correlated with actual field application, if that is what you mean.

And I believe the second question was that you want some idea of the magnitude of tube noise. I can give you some specific figures. They are much higher than the corresponding values for 15 g, 40 cycle vibration—nearly ten times higher for the type 5840 pentode. The rms value (distribution not normal) has a geometric mean of around 30 mV and the readings for a small sample extend from 5 to 100 mV. The peak-to-peak noise output readings for the same type would extend from 50 to 1000 mV.

W. Bowden, General Precision Laboratory: What kind of voltmeter did you use to measure the rms value?

Robbins: A true rms voltmeter was used to measure rms acceleration. We have a special voltmeter manufactured by Ballantine.

Bowden: True rms?

Robbins: Yes, it is a fairly good sized piece of equipment, about 8" x 8" x 20". For complex waves with peak-to-peak/rms ratios of the order of ten to one or higher, there is a possibility of small error.

I might also mention here, that the noise output of the tube under test should not be considered as white noise. A true rms meter would be required for precise rms readings of tube noise, but a conventional rms vacuum-tube voltmeter of the averaging type is sufficiently consistent and satisfactory for actual vibration testing.

* * *

WHITE-NOISE TESTING OF ELECTRON TUBES

J. M. Stinchfield and R. E. Dorrell
Diamond Ordnance Fuze Laboratories

Field experience in rocket development is discussed along with its extension to laboratory simulation leading to the development of white-noise testing of electron tubes. The testing technique as applied to the design of experimental tubes is described briefly. Results, in acceptance testing and phenomena experienced, are outlined.

In the design and development of electronic devices for use in the field, consideration must be given to both functional and environmental requirements. An electron tube or other electronic device not only has its surrounding inside environment but has also coupled to it reflected components of the outside or the remote environment. A study of the apparatus and the field conditions is necessary to evaluate these factors. With this information in hand the desirability and extent of simulation of these environments when testing the component device can be determined.

It was often suggested in early work on rocket fuzes that a playback of recorded field conditions, as a source of drive for a vibration-generator, would be a satisfactory simulation of field conditions when testing an electron tube. While there may be situations where this technique is of value, it is evident from the following that in this case it should not be so used. There is a large variation in vibration amplitude and frequency spectrum. Not only is this so in change of vibration mode within a given rocket, but in variation from one rocket to another of the same kind. Further, the same electron tube must also be used in more than one type of rocket. Obviously, the matching or displacement of rocket resonances with respect to tube resonances would be very unpredictable. Also, any combination would not represent a particular field condition.

It is also found in the study of electron tubes that noise, microphonics, and shock effects occur over a wide range of frequency and amplitude. Variations from tube to tube made under the same conditions can be appreciable.

It is evident that a range of frequency spectrum and of g level is indicated in the foregoing application for the testing of electron tubes or similar electronic devices. The experience shows that low frequency shaking or bumping is not adequate in this work. The need for adequate excitation of the higher frequencies is evident, plus effects to be described in a later part of this discussion.

The development and use at Diamond Ordnance Fuze Laboratories of an electrically driven vibration-generator (1) (Figure 1) with flat response from 100 to 10,000 cps at the 10 g level, plus the development of a simple accelerometer (2) (Figure 2) has provided reproducible test conditions. It was also found that while many types of excitation can be used with this equipment that the simulation provided by the continuous and uniform frequency spectrum of white noise was most useful as a simple over-all test.

Since the introduction of white-noise testing at these Laboratories, the use of white noise as a source of excitation has found increasing use at other laboratories (3,4) and is now considered

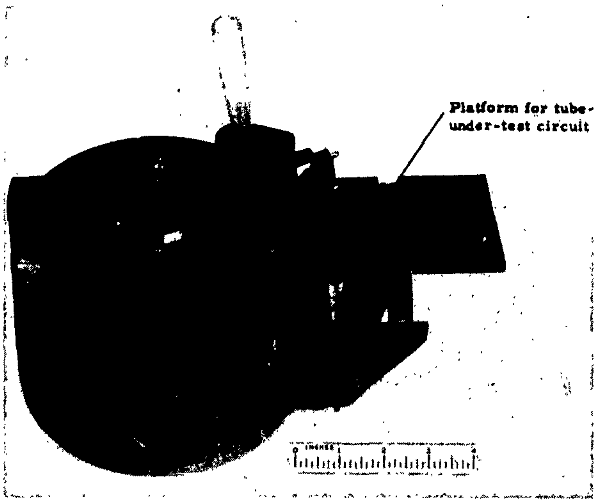


Figure 1 - Vibration-generator showing T-3 tube inserted in armature

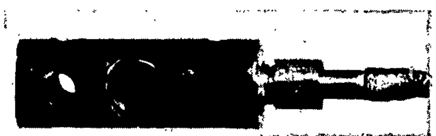


Figure 2(a) - Subminiature tube type accelerometer

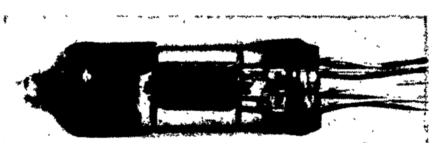


Figure 2(b) - Subminiature tube (T-3 envelope)

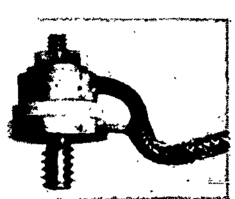


Figure 2(c) - Model M-3 accelerometer

one of the tools for the microphonic investigation of tubes. In this paper the testing technique and some results in the application of this method to fuze tube development are considered.

Figure 3 shows the complete test equipment including the white-noise source, the noise-power amplifier to drive the vibration generator, the tube under test, voltage supply, and the vibration output indicating devices.

Calibration procedure is as follows: A barium-titanate accelerometer having a sensitivity of about 2 millivolts per g of acceleration and of the same weight and size as the tube under test is used.

The white-noise source is replaced with a sine-wave 1000-cps source.

The rms value of the 1000 cps current to the moving coil of the vibration-generator is increased until the required acceleration is read.

The 1000-cps source is replaced by the white-noise source and the rms current to the moving coil adjusted to the same value as was read with the 1000-cps source. This corresponds to the same total energy in both cases.

This completes the calibration. The calibration is simple and is retained for long periods.

As applied to the design of experimental tubes the output of the tube-under-test goes to the cathode-follower stage and output-indicating device with broad-band characteristics. In this way either the white-noise source as described above or an exploring sine-wave source may be used. Any frequencies excited by the source can be observed within the limits of the broad-band output device. Not only does this supply a figure of merit for the tube over-all quality but also detailed information on the frequency location of resonances and the probable sources of these.

From its inception, early in 1941, the V.T. Fuze employed commercial hearing aid tubes and as time progressed these tubes were refined to meet the needs of the field. Early tubes consisted of a single filamentary emitter that was tensioned by a helical spring and supported by the tube anode. Later, a second filament and its associated spring was added to increase the emission of the device. These filaments were both strung over mica straight edges and held taut with a spring tension of ten to fifteen grams.

The filaments were enclosed in the evacuated glass envelope of the tube and possessed an extremely high Q. The value of Q for a typical tube ranges from 5 to 10,000. At the time the electrical "improvements" were being made, it

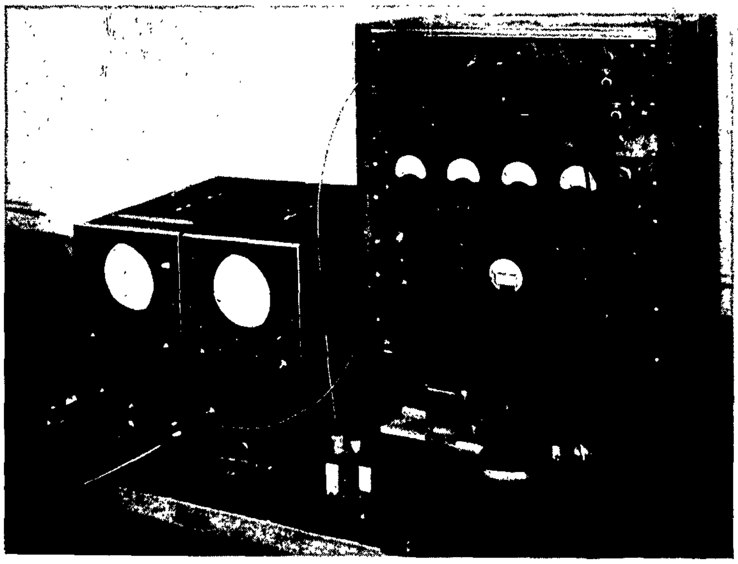


Figure 3 - TN1 vibration test setup

was discovered that microphonics troubles had been increased also. This trouble was discovered by poor field scores, but little or no laboratory correlation could be discovered. It was generally believed that the two filaments within the tube, each with a distinctive frequency, were giving rise to the increased microphonics by a beating or heterodyning action which, in turn, produced a low-frequency signal in the passband of the Doppler-frequency amplifier. It was in the attempt to verify this generally accepted theorem that white-noise testing was born at D.O.F.L. in 1949.

One of the first requirements in testing the filament intermodulation theory in the Laboratory was that both filaments had to be excited simultaneously. This was not difficult as long as looseness existed in the tube because one rattling tube element will shock-excite others. The tubes were developed to such a state that all element looseness had been "engineered" from the mount and a sturdy, rigid structure was available for testing.

The first difficulty that arose was that it was virtually impossible to excite both filaments simultaneously. In the laboratory work, forty- and sixty-cycle sinusoidal excitation was used, but due to the freedom from looseness, the filaments were not excited. Square-wave and rectangular-wave excitation was also tried, but due to the finite and discrete nature of the spectrum of each of these exciting media it was sheer accident if both filaments were excited.

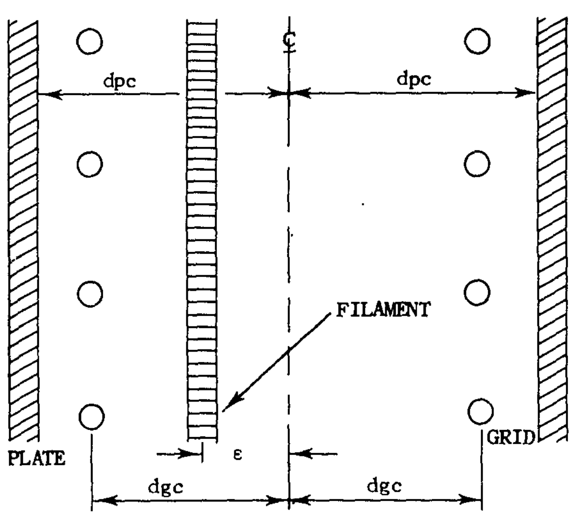
It was discovered quite early that tapping the tube with a pencil or cork mallet would accomplish the end result but a mechanical device to perform this function was not then available. Hand tapping was very difficult; to analyze a microphonic spectrum while tapping a tube was next to impossible. Another method tried was to connect two signal generators in series, with each one tuned to a filament resonance. This method did excite both filaments but was unsatisfactory because the electrical and mechanical system of the vibrator generated a difference frequency due to associated nonlinearities. As a desperation measure, a noise modulator from an APT-5 radar jamming transmitter was tried and it worked extremely well. The wideband noise excited not only filament resonances but resonances in the loudspeaker-type vibrator then being employed and led to the development of the well-known N.B.S. flat vibrator.

After the struggle to excite both filaments simultaneously was concluded, the main test work was continued. In a very short time it was proven that filament intermodulation frequencies were not to blame for the microphonic troubles and that proper tube design could eliminate the small amount found to be existing. A second verification of this was obtained when it was noticed that single filament tubes were equally as microphonic as the two filament variety. It was definitely proven however that tube microphonics was associated with the filament. This was brought out as follows. A band-rejection filter was inserted between

the noise source and the amplifier driving the voice coil. When this filter was tuned to eliminate the filament frequencies, all microphonic noise vanished. It was then decided to continue using white-noise excitation for tube testing purposes in order to make certain that all tube resonances would be excited simultaneously.

The microphonics problem was then approached from a theoretical standpoint with the analysis proceeding as follows:

In a two-sided single-filament tube, assume that the grid and anode are planar and are spaced equidistant, on each side, from the centerline of the tube structure. The filament is considered to be eccentric or displaced from the centerline as shown in drawing.



The anode current for a filamentary triode is given by (5)

$$I_b = \frac{2.33 \times 10^{-6}\, A}{d_{gf}^2} \quad \frac{[E_c + E_{b/\mu}]^{4/3}}{\left[1 + \frac{1}{\mu} + \left(\frac{4}{3}\right)\left(\frac{1}{\mu}\right)\left(\frac{dgp}{dgf}\right)\right]^{4/3}}$$

or by simplifying,

$$I_b = \frac{K}{d_{gf}} \text{ amperes.}$$

Assume: Filament vibrates with amplitude αf and angular rotational rate ωf about its static position shown, the displacement being given by $af \operatorname{Sin} \omega ft$.

Similarly, the grid displacement is given by $ag \operatorname{Sin}(\omega gt + \phi)$ where ϕ is an arbitrary phase angle. The anode is assumed stationary. Summing the current to both anodes (as they are joined electrically)

$$I_b = K \left\{ \frac{1}{dgc - \varepsilon + af \operatorname{Sin} \omega ft - ag \operatorname{Sin}(\omega gt + \phi)} + \frac{1}{dgc + \varepsilon - af \operatorname{Sin} \omega ft + ag \operatorname{Sin}(\omega gt + \phi)} \right\}.$$

This current is really an increment along the length of the filament, and to obtain a quantitative answer, an integration should be performed along the length of the tube. This results in a constant expression and was not done here as the point of interest lies in the qualitative expression. Expanding and using the first three terms of the Binomial expansion, the following terms of interest result:

Term	Coefficient
dc (Steady)	$\left(\frac{K}{dgc - \varepsilon}\right)\left\{1 + \frac{1}{2}\frac{af^2 + ag^2}{(dgc - \varepsilon)^2}\right\} + \left(\frac{K}{dgc + \varepsilon}\right)\left\{1 + \frac{1}{2}\frac{af^2 + ag^2}{(dgc + \varepsilon)^2}\right\}$
Sin ωft (Filament Fundamental)	$K\, af \left\{\frac{1}{(dgc + \varepsilon)^2} - \frac{1}{(dgc - \varepsilon)^2}\right\}$
Cos $2\omega ft$ (Filament 2nd Harmonic)	$-\frac{K\, af^2}{2}\left\{\frac{1}{(dgc + \varepsilon)^3} + \frac{1}{(dgc - \varepsilon)^3}\right\}$
Sin $(\omega gt + \phi)$ (Grid Fundamental)	$K\, ag \left\{\frac{1}{(dgc - \varepsilon)^2} - \frac{1}{(dgc + \varepsilon)^2}\right\}$
Cos $2(\omega gt + \phi)$ (Grid 2nd Harmonic)	$-\frac{K\, ag^2}{2}\left\{\frac{1}{(dgc - \varepsilon)^3} + \frac{1}{(dgc + \varepsilon)^3}\right\}$
Cos$\{\omega ft - (\omega gt + \phi)\}$ Lower Intermodulation Frequency	$-K\, af\, ag \left\{\frac{1}{(dgc - \varepsilon)^3} + \frac{1}{(dgc + \varepsilon)^3}\right\}$
Cos$\{\omega ft + (\omega gt + \phi)\}$ Upper Intermodulation Frequency	$K\, af\, ag \left\{\frac{1}{(dgc - \varepsilon)^3} + \frac{1}{(dgc + \varepsilon)^3}\right\}$

The following can be concluded from the tabulation. The dc anode current should increase if the filament is excited and eccentric. This agrees with laboratory observations. The filament and grid fundamental frequency should vanish if the filament eccentricity is zero. This case has not been observed experimentally, leading one to conclude that if the conditions assumed are valid it is extremely difficult to construct a tube with a perfectly centered filament.

The term of especial interest is the Lower Intermodulation Frequency as it is capable of falling in the passband of the Doppler-frequency amplifier. Note that its amplitude is proportional to both filament and grid amplitude so that a large amplitude (high Q) filament disturbance can interact with a much smaller amplitude grid resonance and produce a microphonic signal capable of failing a tube. With white-noise excitation both the filament and grid are excited simultaneously and the intermodulation results. It is believed that a burning rocket motor supplies the same type excitation.

By employing a Sonic Spectrum Analyzer and white-noise vibration excitation this process was demonstrated. A motion picture of the action has been made. The improvement in fuze tubes due to the separation of the tube and filament resonances has resulted in an improvement in microphonic noise by a factor of approximately forty times.

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IMPULSE VS STEADY-STATE EXCITATION IN THE EVALUATION OF ELECTRON TUBES

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The relative effectiveness of impulse vs steady-state excitation in the field of resonance and vibration testing of electron tubes, are compared using a precision impulse exciter. Accelerometer techniques for deriving impulse spectra from this exciter are described. Optimum excitation, representative of the broadest range of field environments, is considered.

INTRODUCTION

This age of jets and guided missiles has witnessed an increase in the severity and variety of mechanical environments for equipment in military applications. Among the more important resulting changes has been the extension of the frequency range of mechanical excitation to which electronic components are subjected, and the realization that much of this excitation is transient and aperiodic in nature.

Large-scale programs have been launched to provide equipment and components with superior mechanical properties, capable of withstanding these harsher conditions. This in turn imposes a necessity for corresponding advances in the procedures for testing these properties. Improved testing procedures are urgently required to serve a fourfold purpose:

- Provide a critique for assessing the actual degree of improvement achieved by development engineers,
- Provide a quality control tool for maintaining the improvement during production,
- Enable the preparation of valid and realistic specifications, which can ensure uniformity of quality irrespective of the supplier,
- Provide a degree of predictability as to performance in actual field environment.

The studies being reported herein originated as program to improve the techniques for evaluating the mechanical characteristics of electron tubes. However, the philosophy generated, and some of the results achieved, may have more general significance in the field of mechanical testing for those structures which behave in a manner similar to electron tube structures.

A closer look at these structures may help demonstrate this applicability. The electron tube contains many elements, combined in a very complex structure, which interact as multiple-degree-of-freedom, coupled oscillators. The operation of the tube is dependent on the relative position of the elements, so that when mechanical excitation displaces them, the electron tube generates a spurious electrical signal or noise, which if severe enough, may represent malfunctioning. The tube behaves as a transducer of mechanical into electrical energy. If the electrical output is large, due either to looseness or low damping, and the mechanical excitation continues over a long period of time, a cumulative fatigue damage may occur. Thus the electron tube may be considered as an active device which tells you what is happening to it in some detail, and also whether it hurts! In delivering this information, it differentiates itself from the conventional passive structure. However, the susceptibility to fatigue is common to both types.

In the present Basic Section of the MIL-E-1B Specifications for Electron Tubes, provision is made for tests to determine the tendency of an electron tube to generate spurious electrical signals in the audiofrequency band, when mechanically excited. The excitation employed in one test (1) is obtained by striking the tube envelope with a cork hammer, wielded by hand. Another test (2) employs an automatically operated hammer, which delivers its blow indirectly to a chassis on which an electron tube is mounted in a socket. In both cases, the repeatability and reproducibility are unsatisfactory, the frequency range of the mechanical excitation is quite limited, the electrical detection apparatus operates over a restricted frequency band, and the rejection criteria are subjective and inadequate.

THE NML SYSTEM

In attempting to improve these tests, the Material Laboratory has devised a system intended to overcome these limitations. A new technical exciter was designed, into which the following general properties were incorporated:

1. As sharp an impact as reasonably attainable, to provide widest possible band width of excitation frequencies.

2. Delivery of the impact directly to the envelope of the tube under test, thereby shortening the transmission path to the tube mount to an irreducible minimum.

3. The use of gravity as a driving force, to remove the need for calibration, to ensure excellent repeatability of results, and to achieve reproducibility of the blow at all installations.

4. Fabrication of the critical members from readily available materials with closely specified mechanical properties which do not change with time or use, again to avoid complications of recalibration procedures.

5. Automatic operation, to enable use in quality control and production testing.

This equipment, which has come to be known as the NML Standard Pendulum Tapper, is shown in Figure 1. The basic element of this exciter is a rigid pendulum of precisely specified dimensions, materials, mass, and distribution of mass, utilizing a hardened steel bearing ball as the bob and impacting body. This device is inherently capable of delivering a large force of extremely short duration. It is interesting to note that this is not the first time a pendulum has been employed to excite the audiofrequency noise output of tubes. Some time after the development of this tapper, our attention was called to the early work of Alan Rockwood and W. R. Ferris (3), who utilized a pendulum device in their study of microphonism.

Referring now to the numbers in Figure 1, the bob (1), a steel bearing ball of 1/2-in diameter, is affixed to a "Y" made of drill rod (2). The choice of steel for the ball was made to obtain as sharp an impact as possible. The upper arms of the "Y" are pivoted on ball bearings (3), housed between two horizontal bars (4), which are supported rigidly by two end frames (5). An adjustment (6) allows the pendulum arm to be translated horizontally over a small range in order to bring the bob in its zero position to the point where it just makes physical contact with the envelope of the tube under test. Thus, all possible diameters of tube envelopes may be accommodated.

The base plate (7) supports the socket (8) into which the tube under test (9) is inserted. The tube is held only by the socket clips. It has been found that variations in the socket support have negligible effect on the blow delivered. Vibration isolation elements (10) in each corner suspend the base plate in such a manner that no extraneous excitation can reach the tube under test from either the superstructure or the table on which the equipment stands.

The adjustable feet (11) permit the raising or lowering of the bob relative to the tube under test. In addition, an adjustment (12) is provided for rotating the tube socket about its vertical axis with respect to the base plate. Thus, any predetermined point of impact on the tube envelope may be struck. The base plate and pendulum support frame may be levelled (13), so as to bring the rest position of the pendulum to indicated zero degrees. The pendulum may be released at any desired angle along the sector arm (14), which is graduated in five degree intervals. The release mechanism (15) is a simple spring-actuated retraction device, which imparts no momentum in any direction to the pendulum.

In designing this exciter, an effort was made to obtain an absolute instrument in the sense that no external reference standards would be needed for the purpose of calibration except those of mass, length, and time. This was achieved by utilizing gravity as the driving force, by careful choice of materials, and by design such that variation in dimensions within

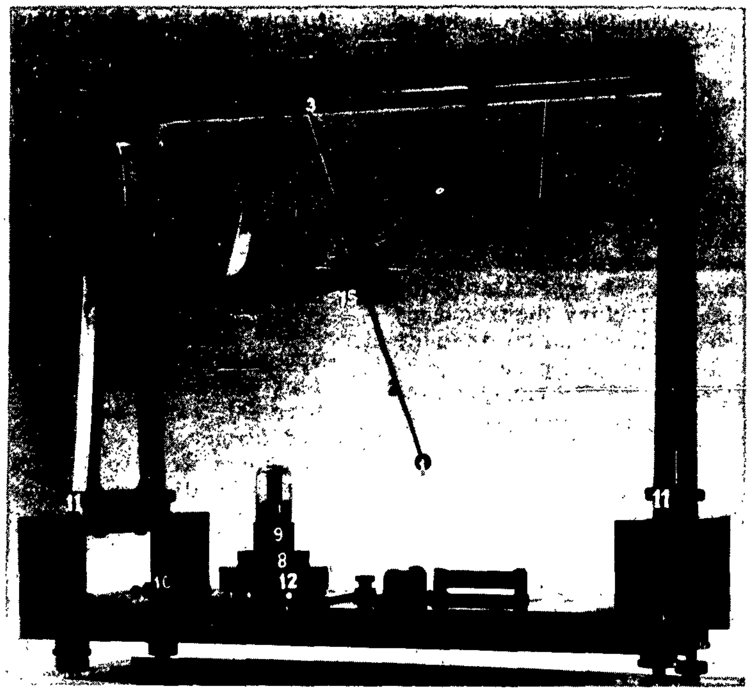

Figure 1 - NML standard pendulum tapper

normal machining tolerances would yield negligible effects. At this point, it is perhaps necessary to state that no instrument of this type can be calibrated in terms of delivered impulse, since the latter quantity is dependent upon the mass and elastic properties of the object struck (which vary from tube type to tube type), as well as the properties of the instrument. At best, one can obtain a calibration of momentum or energy available immediately before impact, or their equivalents. In a pendulum, the momentum before impact is directly proportional to the angle of release. Hence, the calibration is accomplished in manufacture, so that no further calibration procedure is necessary.

In order to determine whether the specifications on mass and the distribution of mass in the pendulum have been adhered to in manufacture, and to insure the necessary degree of cleanliness of the ball bearings, two operational checks are required. A period measurement and a decrement measurement, employing as the reference standard the second hand of an ordinary watch, fulfill all the necessary operational safeguards required of the instrument to insure its precision performance. Although, not shown in this Figure, a motor and friction cam drive have been added to the tapper to obtain automatic operation. When the open section of a rubber driving cam opposes the bearing housing at (3) in Figure 1, the pendulum falls freely with no encumbrances. The solid portion of the cam rolls around into contact with the bearing housing just in time to pull back the pendulum, preventing it from rebounding and hitting the tube a second time. The pendulum is pulled back until the ball hits a simple stop, the position of which determines the angle of fall. The pendulum arm is then held at this point, against the slipping clutch action of the cam in contact with the bearing housing.

To accompany this precision exciter, a similarly precise indicator is required, covering the entire audioband. Studies of tube output transients when struck by the pendulum tapper indicated that measurement of two parameters would serve to characterize adequately both the spikes of impulse noise, and the damped transients due to microphonism; these two parameters are the peak amplitude and the average (or time-integrated) value. Large values for either of these may produce serious interference with the normal operation of circuits employing the tube. Neither measurement alone is sufficient since, in many cases, no proportionality exists between the two parameters.

The indicator designed for this system includes a low-noise, high-quality audioamplifier,

having a frequency response of 20 to 20,000 cycles. As may be seen in Figure 2, after passing through the amplifier, the transient output of the tube under test is separated into peak and average (integrated) measuring channels and then full-wave rectified.

tubes of the same type, as in production testing. However, it is frequently desirable to obtain quantitative information concerning the peak and average output of each tube, as, for example, in our extensive investigations of tube noise of which a small part will be reported below. For this purpose, oscilloscopes are substituted for the thyratron circuits.

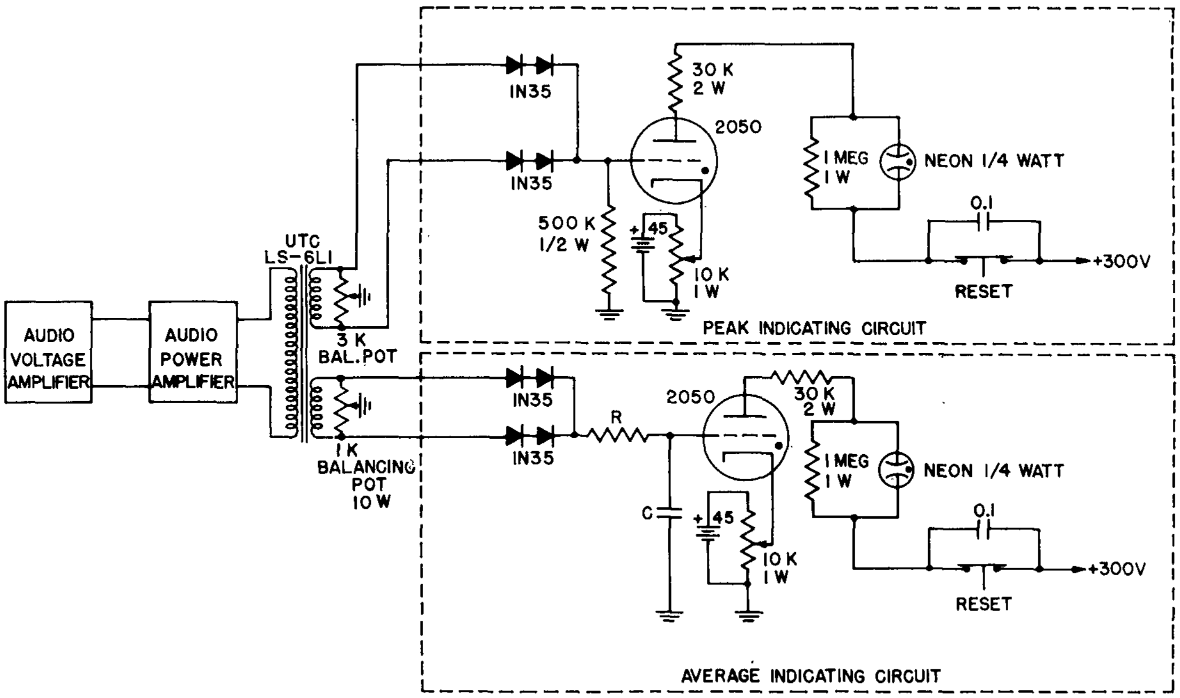

Figure 2 - Indicator schematic

In the peak channel, the rectified output is fed directly to the grid of a thyratron whose bias is adjusted to be below its firing point by an amount precisely equal to the largest acceptable (amplified) peak. Should this level be exceeded, even momentarily, by the incoming noise transient (without regard to the sign of the noise-voltage peak), the thyratron will fire, thereby lighting a neon bulb in its plate circuit. In the integrated, or average channel, the full-wave rectified current is fed to an adjustable R-C integrator, so that the capacitor C is charged through the resistor R. If the voltage accumulated across the capacitor exceeds a preset value, determined by the bias setting on the thyratron, that tube will fire and light another neon lamp. Thus, a quick glance at the two neon lamps (which stay lit until the circuits are reset) present the answer in an entirely objective, "go no-go" fashion.

This type of presentation is particularly convenient for rapid evaluation of large numbers of

Since the intent was to build into the pendulum tapper certain desirable characteristics for the excitation of electron tube noise, how closely these desirable features were achieved was experimentally investigated. Analysis of the frequency spectrum of the mechanical impulse delivered to the tube envelope posed a difficult problem, since an accelerometer could not be mounted on the tube at the point of impact without seriously affecting the actual impulse. However, since the pendulum experiences a force equal but opposite to that exerted on the tube, it was decided to measure the blow reacting on the pendulum ball during impact. The effective masses of pendulum and tube being different in general, the magnitude of the acceleration experienced by the tube is not necessarily equal to that measured on the pendulum ball, although it is readily calculable. For the present, it is

sufficient to know the shape and duration of the acceleration which the tube undergoes.

As shown in Figure 3, a pendulum ball was cut in half and a Massa M142 accelerometer was mounted on the remaining half. The total mass of half the ball and accelerometer was slightly increased over the normal pendulum, while the material and geometry of the impacting surface were unchanged. In order to ascertain whether the blow delivered by the modified pendulum arm was the same as the blow delivered by a standard pendulum arm, each was, in turn, permitted to strike an accelerometer, the output of which was recorded. The two recorded accelerometer outputs were compared and found to be essentially the same.

The modified pendulum arm was next caused to strike three different types of electron tube envelopes: a metal octal, a glass octal, and a miniature tube. The resulting oscillograms of the accelerometer outputs are shown in Figure 4. The shortest impulse (95 microseconds) is generated by hitting a glass octal tube, while the impulse delivered to a metal octal tube is more than twice as long. The miniature tube falls in between these two. The high-frequency modulation is a 50 kc oscillation, ascribed to the accelerometer resonance. Ignoring this modulation, the three waveforms may be approximated by sine functions. The application of Fourier Integral Analysis to these sine functions yielded the spectral content of the blow delivered to the three tube types.

As can be seen in Figure 5, these spectra show constant acceleration at low frequencies, falling to zero at 7500 cps for metal octal,

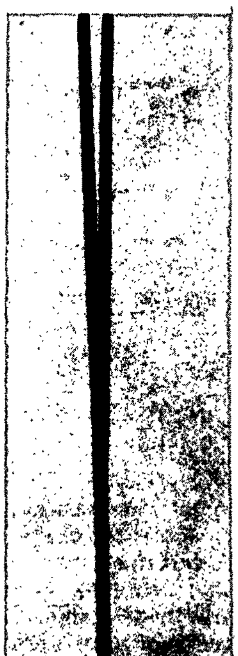

Figure 3 - Accelerometer mounted on modified pendulum

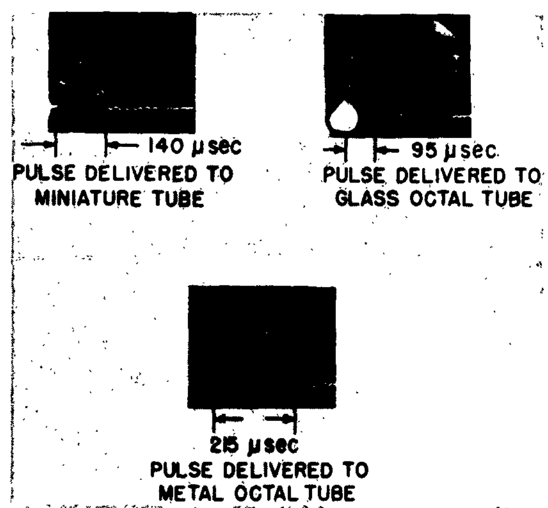

Figure 4 - Mechanical impulses delivered to tube envelopes by the pendulum tapper

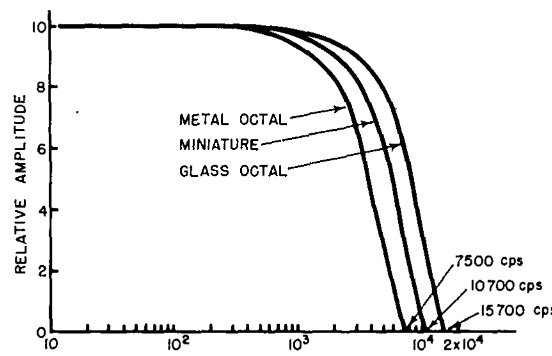

Figure 5 - Graph of frequency spectrum delivered to tube envelopes by the pendulum tapper

10,700 cps for miniature, and 15,700 cps for glass octal tubes. Fortunately, most of the natural modes of vibration lie below these points. In this Figure, the spectra have been shown only as far as the first zeroes; actually, there is appreciable energy extending to much higher frequencies. For convenience, all three curves have been plotted with equal values of acceleration at low frequencies.

Linearity is an important requirement of the system when quantitative comparisons among tubes are desired. This characteristic was examined in some detail. It was found that the momentum of the pendulum just before impact is almost directly proportional to its angle of fall. From simple energy considerations, it can be shown that the departure from linearity at 30 degrees is only 1.2 percent. This was verified experimentally by allowing the pendulum to hit a Sperry-MIT pickup mounted on a tube base plugged into the socket of the pendulum tapper. A plot of pickup output versus release angle is depicted in Figure 6, where each plotted point represents the mean output for 152 blows. This graph is linear between five and thirty-five degrees. Below about five degrees, friction in the bearings produces a noticeable effect.

On an a priori basis, it is clear that this equipment should be inherently capable of excellent repeatability and reproducibility. The repeatability was determined over a period of several years at the Material Laboratory and, with the kind cooperation of the Signal Corps, at the Evans Signal Laboratory at Belmar, New Jersey, using three different tappers independently constructed from the specifications. Numerically, the coefficient of variation turned out to be less than 2 percent. As for reproducibility, comparisons of the results at the two Laboratories revealed a coefficient of variation of about 3 percent. It should be kept in mind that these variations included that due to observers, pickups and oscilloscopes, as well as the pendulum tappers.

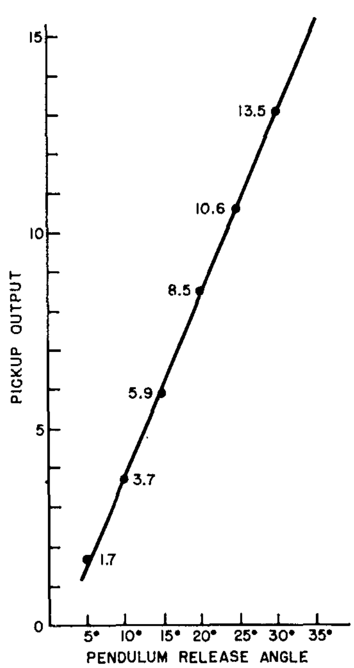

Figure 6 - Graph of pickup output vs pendulum release angle

Having devised an exciter and indicator system which met high standards as to frequency range, objectivity, repeatability and reproducibility, it was applied to an extensive program of study on the impulse noise and microphonic output of tubes, as reported elsewhere (4,5,6) with great success. The Standard Pendulum Tapper, in conjunction with a special wide-band peak indicator circuit (from 20 kc to 10 mc) has been employed for the measurement of mechanically-induced RF noise, and for the detection of shorts and discontinuities in electron tubes, in both cases with excellent results. These signal successes demonstrate the usefulness of this equipment as a quantitative research instrument.

However, there are phases of the applicability of these techniques which may prove to be of more general interest in the broader field of mechanical evaluations, which will be explored now in some detail.

APPLICATION OF IMPULSE TESTING FOR RESONANCE

The formulation of a mechanical resonance test has long been a necessity in order to fulfill the requirements of military specifications, and it was to this field we next turned our attention. The resonance evaluation, as applied to electron tubes, is not defined in MIL-E-1B Specifications; as a matter of fact, the only information one has as to what is expected of the test is that there is to be no pronounced resonance below 100 cycles. As might be expected, there have been many spirited discussions between buyer and vendor as to what is a "pronounced" resonance in a tube, and what is not.

A few years ago, at the time we started our investigation, there seemed to be fairly universal, although tacit agreement that the test

was to be performed by placing the tube under test on a shaker and vibrating it from 10 to 100 cps. The method of detecting the resonance, however, varied from visual observation, with and without microscope and stroboscope, to electrical metering. In the electrical method, the most common technique utilized an oscilloscope or a vacuum tube voltmeter to view the signal developed across a load resistor. As the frequency was swept mechanically through the band, the output was monitored for a peak or maximum, which would indicate a resonance.

Investigation of these techniques showed that visual observation has the advantages of being quite direct and straightforward, and reveals which element is actually vibrating. It is an excellent tool for studying mechanical resonances, but leaves much to be desired as a test, since it is limited to glass envelopes and to certain tube structures, orientations, and sizes of tubes. Furthermore, it is quite time-consuming and requires a fair degree of training to perform.

The peak-output method is applicable to all tubes. However, the maxima are frequently not sufficiently defined, and may be small in magnitude and not easily distinguished from the general noise level. The signal-to-noise ratio may be increased by vibrating the tubes with larger force, but this causes some degree of damage or, at least, permanent change at the resonance points. At best this is a lengthy procedure, and not productive of repeatable results, due to changes in the tubes caused by excitation at the resonant frequencies.

Our studies turned up some other methods with various advantages. For example, there is a phase-shift method which involves feeding a signal from an accelerometer mounted on the shaker table, to the "X" deflection plates of an oscilloscope. The signal across the tube load resistor is fed to the "Y" plates. When the tube is vibrated, a Lissajous pattern is formed on the oscilloscope, which goes through a readily observable phase shift as the shaker is swept through a tube resonance. This is much more sensitive and reliable than the peak-output method. Like the peak-output method, however, it has poor repeatability (for the same reasons), and requires a point-by-point analysis.

Another method, very sensitive but requiring even more tedious manipulations, is to measure periodic changes in interelectrode capacity. For a complete analysis, this requires measurements between all combinations of pairs of electrodes.

It can be seen that the common denominator of all these methods is a lengthy point-by-point analysis which suffers in repeatability because of changes induced in the tubes by vibrating them at their resonant frequencies. Furthermore, there remains in each case the problem of how to define what is and what is not a resonance.

At this point in our studies, a question arose, engendered by the fact that we were engaged concurrently in investigating the a-f microphonism test. Since a-f microphonism, as revealed by the pendulum tapper, is a result of the excitation of the tube resonances, why not use the same system for a resonance test?

Here a question of philosophy intervenes. In all the methods of resonance testing discussed above, there is what may be called a diagnostic feature; namely, one learns at what frequency the resonance occurs. However, when a tube is failed by the pendulum tapper and associated circuitry, all that is known is that the output of the tube exceeds a certain value. Is the diagnostic feature necessary?

We believe that as a consumer of tubes, the Government is not interested in diagnosing the reasons for failing. Its principal interest, we further believe, is to have available means for reliably separating the wheat from the chaff in as accurate, objective, expeditious and inexpensive a manner as is possible. Let us examine now how well the pendulum tapper system does this for resonance.

There were some questions as to whether the tapper blow would excite resonances below 100 cycles. A good deal of searching (using the visual-observation and the phase-shift methods), finally turned up a tube which had a resonance in this range. It turned out to be an 838 transmitting tube. Parenthetically, our lack of success in finding a receiving tube with such a resonance raised grave doubts as to the suitability of this upper-frequency limit. At any rate, we were able to excite the only resonance in this tube below 100 cycles (namely, at 76 cycles) with the tapper, just as well as with steady-state excitation. This is certainly to be expected, considering the frequency spectrum introduced into the tube envelope by the tapper, which was shown in Figure 5. As may be seen in that figure, the spectrum is flat over most of the audioband and right down to dc.

It was decided at this point to consider 600 cycles as the frequency limit below which there should be no resonance. This was in harmony

with the desires of the Air Force at that time, and also better reflected our thinking than the 100 cycle limit previously employed.

An experimental correlation was attempted now between the tube resonances excited by forced vibration and by tap excitation. Twenty-four tubes from Navy stock were obtained, of three different types: the 6AC7W, 6SL7W and the 5932. Their resonances were determined by sweeping the frequency of an electrodynamic shaker, on which each was mounted in turn, through a band from 20 to 800 cycles. Resonance was detected by both the peak output and the phase shift method; the output maxima and the frequencies of the various resonances were noted.

A correlation was originally desired between these output maxima determined from vibration and the resonant amplitudes determined from tapping. However, repeated trials of the tubes on swept-frequency vibration showed such a variability in the output at resonance that this attempt was abandoned. Instead, it was decided to compare the actual frequencies at which resonances were recorded, since these repeated very well.

In order to determine the frequency components present in the transient output of the tubes excited by the pendulum tapper, the output was recorded on a very short, endless magnetic tape, and played back repeatedly into a frequency analyzer, so that the output at each frequency could be measured. The high degree of correlation obtained may be seen in Table 1 where essentially every resonance revealed by the steady-state, or vibration method, was also shown by the tapper-impulse method. The impulse method excited additional resonances, which will be discussed later.

As shown in this table, the differences between the two frequency measurements in each case are expressed as a percentage. The incidence of these percentages was plotted against percentage difference, and a Gaussian distribution resulted. This showed that the differences were not significant, but could be attributed to experimental error. Thus, since the same resonant frequencies are excited by impulse that are excited by sinusoidal excitation, then impulse may be substituted for steady-state vibration to excite the mechanical resonances of electron tubes.

Since these two methods are, at least to this extent, interchangeable, one must judge between them on the basis of other considerations. The

TABLE 1
Material Laboratory Data
Frequency Correlation for Steady State and Impulse Excitation

Tube No.	Freq. Excited Impulse Exc. (cps)	Freq. Excited Steady State Exc. (cps)	% Diff.
1	265 630	275 600	-3.7 +4.9
2	300 825	280 840	+6.9 -1.8
3	315	330	-4.7
4	285		
5	295 610	310 600	-5.0 +1.7
6	315	325	-3.1
7	280	280	0
8	280	285	-1.8
9	320	330	-3.1
10	295	260	+13
11	305 585	305 585	0 0
12	350 640	330 680	+5.7 -6.1
13	275	280	-1.8
14	245 392 585	 365 540	 +7.1 +8.0
15	240 345 475	240 340 480	0 +1.5 -1.0
16	114 255 355	117	-2.6
17	410 190	395 540	+3.7
18	470 298	415	+12
19	145 195	150 205	-3.4 -5.0
20	155 320	156 335 195	-0.6 -4.6
21	145 165 280 470	140 180 310	+3.5 -8.7 -11
22	165 205 330	166 205 330	-0.6 0 0
23	145 185 260 320	143 190 250 315	+1.4 -2.7 +3.9 +1.6
24	140 195 250 320	135 185 230 320	+3.6 +5.3 +8.3 0

advantages of the impulse or tapper method may be enumerated as follows:

- A very simple criterion exists for the rejection of tubes for resonance: namely, the voltage swing developed by the resonance.
- By the use of the thyratron indicator previously discussed, this criterion may be applied completely objectively.
- The NML Pendulum Tapper System provides a highly repeatable evaluation.
- The tapper is readily reproduced anywhere and does not require calibration to correlate results at various installations. It is much less expensive than wide-band, swept-frequency equipment.
- It is completely nondestructive: tubes are not changed at all by the evaluation.
- It is much more rapid, since the answer for all resonances in the chosen band is obtained instantaneously, as the result of one blow.
- Untrained personnel can operate it and get correct answers.
- It is very simple to change the frequency band of interest, simply by changing a filter setting. If desired, it could be extended to one, five, or even ten kilocycles.

In return for this impressive list of advantages, the only disadvantage is that it is not diagnostic, which, as stated previously, the consumer can forego with no loss. A tremendous gain is that, with such simple and rapid procedures, it may serve as an excellent production test.

APPLICATION OF IMPULSE TESTING FOR VIBRATION

Having taken full advantage of the common elements between a-f impulse noise and microphonism, and resonance, it seemed natural to turn attention to a closely related evaluation, namely, the operational vibration paragraphs in MIL-E-1B. These tests subject the tubes to sine-wave excitation, and reject on the basis of excessive voltage output. Paragraph 4.9.19.1 refers to low-frequency vibration (at 25 cycles), 4.9.19.2 is high-frequency vibration (that is, 50 cycles), paragraph 4.9.20.3 is variable-frequency vibration (from 10 to 50 cycles), while paragraph 4.9.20.4 is the same as 4.9.19.1, only this time it deals with a design test.

The variable-frequency vibration test is the most general of these, therefore most of the comments will be directed to that. Looking, as we did before, at the objections to present techniques, the following list is readily compiled:

- Low-frequency excitation is employed, giving little information concerning the tubes at higher frequencies, whereas presently available information shows field excitation extending into the thousands of cycles. This deficiency is becoming more widely recognized, to the end that swept-frequency vibration tests up to 10,000 cycles are coming into use. This is very definitely a real advance. However, material will be presented which will raise questions as to the adequacy of steady-state or swept-frequency techniques in general. For the present, let us return to the low-frequency vibration tests.
- For all receiving and even moderate-size transmitting tubes, practically all resonances exceed 100 cycles. Since it is only at or near resonance that large excursions, and therefore, large outputs, are produced, it is mandatory that an evaluation of the operation of the tube above 100 cycles be performed.
- Many tubes will fail a vibration evaluation on one type of vibration machine (a positive-drive type, for example) while passing on another (the leaf-spring or electrodynamic type, for example). This is partly due to inherent tube properties and partly to machine malfunctioning, such as mechanical "slap", introduced by wear or poor lubrication. Mechanical drive machines are cumbersome, expensive, difficult to maintain, and yield results which are not reproducible. Nevertheless, they are widely used, particularly for the heavier tubes.
- Experience with the present procedures have indicated their inability to weed out defective tubes properly. For example, at the Material Laboratory, less than one percent of receiving tube type approval samples, subjected to this test over a period of years, failed because of excessive electrical noise on the vibration test. Yet many tubes purchased from the same manufacturers have become vibration failures in the field.

Now, in contradistinction to these difficulties, the application of steady-state vibration over a wide frequency range, or alternatively, impulse excitation, can excite all mechanical resonances, and also induce the "slapping" of parts in tube assemblies, due to large excursions at resonance. The wide-band, steady-state method involves a laborious point-by-point check, and requires equipment, for the heavier tubes, which is not even available at this time. Furthermore, on the small tubes, for which equipment is available, the results are not repeatable, as was shown in the work on resonance.

Impulse methods, on the other hand, enables instantaneous excitation of all frequencies over a broad band. It is equally applicable to the large as well as the small tubes, and is repeatable and objective. What is of great importance, also, is the fact that all resonances are stimulated simultaneously.

When all possible resonant modes of a complex-coupled, mechanical system (such as an electron tube) are excited simultaneously, intermodulation frequencies may be produced due to mechanical and electrical nonlinearities. Thus, increased output is produced at frequencies which the swept-frequency method cannot possibly excite, since it lacks simultaneity.

The consequences of this are twofold. Where you want to know the worst performance a tube can deliver, under general field type of conditions, it is necessary to impulse-excite that tube. This was graphically demonstrated by the correlation study which has been described under resonance, and shown in Table 1. As was mentioned before, the impulse excitation produced output at frequencies not excited by the swept-frequency method. It is to be expected that in some instances these intermodulation products may yield as high or higher displacement, and therefore electrical output, than are obtained elsewhere in the spectrum. Thus, where this occurs, one could recognize such tubes as rejects only by means of impulse excitation.*

Furthermore, environments in the field contain transient, or impulse excitation, as well as steady-state vibrational excitation. When you try to recreate these in the laboratory, you find that a sharp impulse, containing a very broad band of frequencies, can simulate the effects of both impulse and steady-state excitation, as has been proved and will be proved again by a different method. However, the converse is not true, since steady-state excitation can only simulate itself. It cannot simulate impulse, because it lacks simultaneity. This fact makes of impulse excitation the more general type, and should be weighed in considering comparative merits of the two methods.

At this point, it was decided to determine whether impulse excitation could be substituted for the whole complex of operational vibration paragraphs. It was our belief that none of the MIL-E-1B vibration paragraphs could reveal as much about a tube's electrical noise behavior as could be extracted from it by exposure to the Material Laboratory system.

The problem of determining whether it was justifiable to substitute the pendulum tapper system for vibration was attacked on several fronts. One of the points that had to be proved was that the tapper system measured a parameter that was the same as, or closely related to, that which vibration measures. If such were not so, then a gap would be left in the information obtained from the tubes, if the vibration test were to be eliminated. The procedure decided upon was to obtain the correlation between readings taken on the same tubes with the two methods.

However, before we proceed with this correlation, a great difference in kind between the two methods must be considered and taken into account. The MIL-E-1B vibration paragraphs we were studying excited tubes only up to 50 cycles, while the tapper excites tubes over a very broad band of frequencies.

Many tube engineers were of the opinion, despite its limited frequency range, that the low-frequency vibration somehow revealed information about the operation of the tube at higher frequencies. To investigate this, 32-type 6AC7 electron tubes were picked at random from Navy tube stock and vibrated at 25 cycles and 2.5 g. A low-pass variable cut-off filter was switched from 100 cycle to 20 kilocycle cut-off while the tube was being vibrated. Thus the two bands in which the output was measured was 10 to 100 cycles and 10 cycles to 20 kilocycles. The results of this experiment are shown in histogram form in Figure 7. As can be seen

*It may be seen in Table 1 that there are two instances where a resonance was detected by vibration which was not detected by impulse excitation. This is undoubtedly due to instrumental difficulties; that is, the more sensitive phase-shift method was employed with vibration, whereas the less sensitive peak-output method had to be used with the impulse-excited transients. Furthermore, the necessity for splicing the endless magnetic type, as well as switching transients, raised the noise level when examining the impulse-excited transients. Lastly, the inherent tube nonrepeatability under vibration may have been a contributory factor.

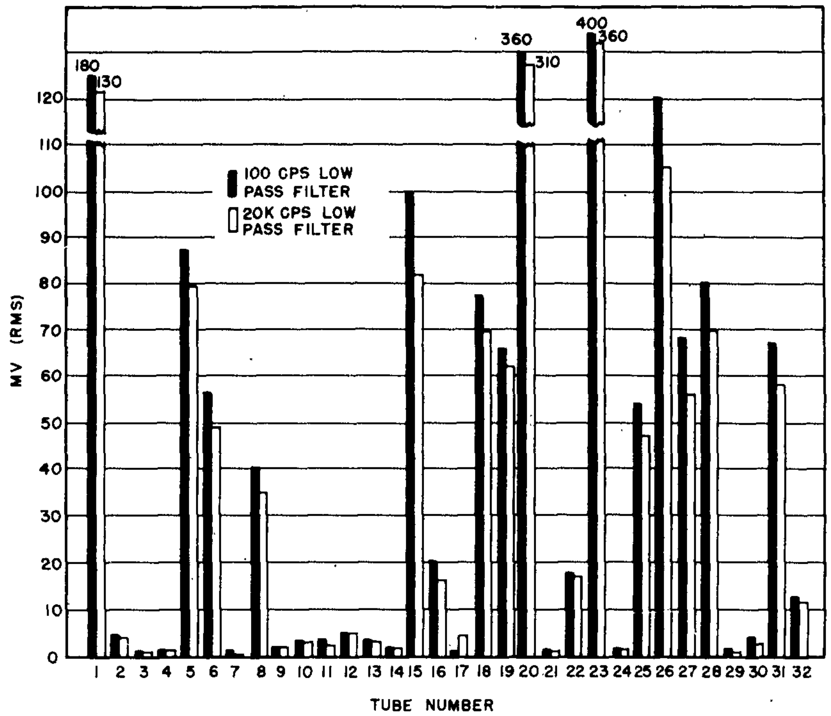

Figure 7 - Comparison of vibration readings (rms) taken with a 100 cps filter and a 20 kcps filter

graphically, the correlation between the outputs in the two bands is close to perfect.* Thus, the MIL-E-1B vibration test yields practically no additional information in the frequency range between 100 cycles and 20 kilocycles, as compared to that obtained from 10 to 100 cycles.

The same type of experiment, with the same two filter settings, was performed on these same tubes with the pendulum tapper. The results for the peak channel are shown in Figure 8. There is very little correlation evident between the outputs taken with the two filter settings. Note the much greater output level observed in almost all tubes with the wide-open band as opposed to the narrow band.

The results for the average channel are shown in Figure 9. Here, no correlation between the two filter settings may be seen. This shows that, unlike vibration excitation, the tapper reveals a great deal of information in the frequency range from 100 cycles to 20 kilocycles, which was not available in the lower band. The broader conclusion must be drawn that a low-frequency vibration test provides little knowledge concerning actual service usage of electron tubes, the pendulum tapper impulse excitation provides a great deal of such knowledge, since transient and vibration excitation at frequencies well above 100 cycles are commonly encountered.

With the information gained from the foregoing experiments, we were prepared to determine the correlation between the two methods. It is now apparent that, to expect any degree of correlation, the tapper system must be degraded frequency-wise to the narrow band to which the vibration tests are confined. This was achieved by using a low-pass filter on the output of the tube excited by the pendulum tapper. As for the vibration output, it makes no difference whether the broad or narrow band is used.

Peak and average tap measurements and three runs of 25-cycle vibration measurements were made using 46 type 6V6GT electron tubes, with the tapper output being degraded by the use

*It may seem strange that the output in the narrow band is greater than that in the broad band, in practically all cases. This is an instrumental defect, in that the insertion loss of the filter was marked as 0 db in both cases, whereas this was not actually so. However, no correction had been made on the original data.

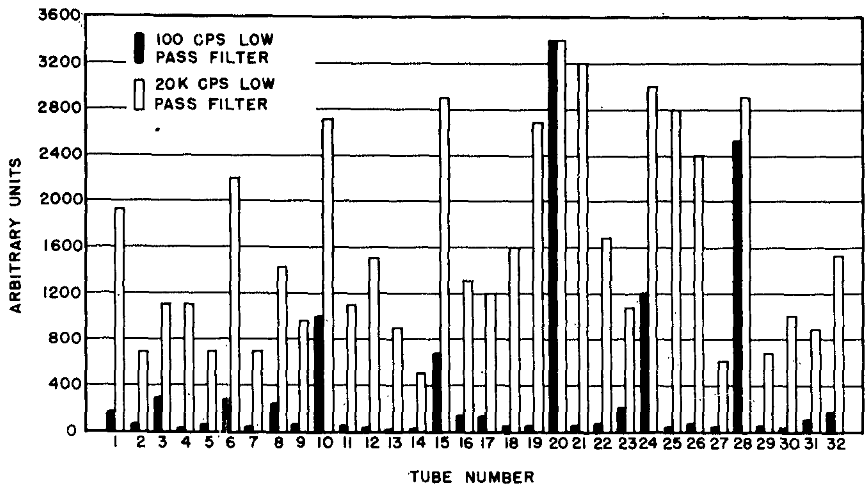

Figure 8 - Comparison of tap readings (peak) taken with a 100 cps filter and a 20 kcps filter

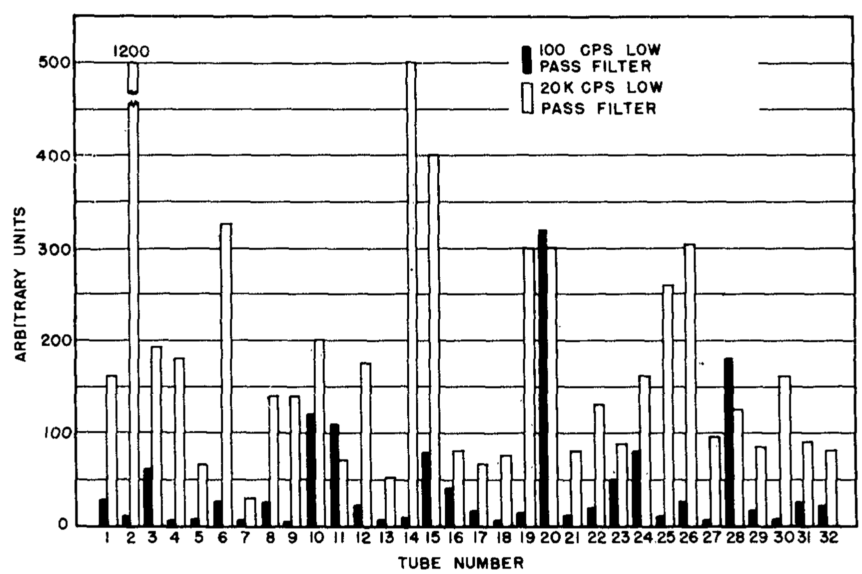

Figure 9 - Comparison of tap readings (average) taken with a 100 cps filter and a 20 kcps filter

of a 50-cycle cut-off, low-pass filter. These data were then treated statistically to obtain product moment correlation coefficients between peak and vibration, average and vibration, and between individual vibration runs. These self-correlations were required because of the poor repeatability of the vibration measurements, as will be seen.

The correlation coefficients of runs 1, 2, and 3 of vibration with the tap average output were 0.56, 0.56 and 0.62, respectively. The correlation coefficients of runs 1, 2, and 3 of vibration with the tap-peak output were 0.40, 0.47 and 0.48, respectively. Now these do not seem to be high correlations. However, this is only a part of the story. The self-correlation of vibration was found to be low also, due to poor repeatability: the coefficients for runs 1 vs 2, 2 vs 3, and 1 vs 3 were 0.68, 0.56 and 0.64, respectively. It may be easier to visualize these relationships by use of data shown here:

TABLE 2

	Tap Average	Tap Peak	Vib_1	Vib_2	Vib_3
Vib_1	.56	.40		.68	.64
Vib_2	.56	.47	.68		.56
Vib_3	.62	.48	.64	.56	

It can be seen that the tap average readings, for example, correlate very nearly as well with vibration as vibration correlates with itself. Another way of looking at this result is to recall one of the meanings of the correlation coefficient, which is: the predictability of the values of one variable from knowledge of the values of the other. Thus, the analysis shows that the results of a tap test on a group of electron tubes may be used to predict the vibration output of these tubes very nearly as well as an actual vibration run, demonstrating the really high correlation between these two tests.

Dissatisfaction with the even-poorer-than-expected repeatability of vibration led to a repetition of this experiment, using 32 type 6AC7 electron tubes, a better shaker, and this time a 100-cycle cut-off, low-pass filter. The self-correlation of vibration was improved somewhat, and the cross-correlation also improved correspondingly. The results, however, were substantially the same, proving as before a high degree of relationship between the two methods.

In order to prove the case beyond any shadow of doubt, to convince any possible remaining skeptics, another approach, which we call operational, was taken with the same data obtained for the second correlation experiment. By operational approach is meant the employment of each test (on a comparable basis) in the manner in which it would be used ultimately; namely, to reject unsatisfactory tubes. Degrading the tap test to include (to a limited extent) only that frequency band involved in the vibration test, should enable the rejection by the tap test of every tube rejected by the vibration test, if we are to substitute the former for the latter. Furthermore, if we have a better test, there should be some tubes rejected by the tap test which are not rejected by the vibration test, thus making the converse impossible; that is, it would not be possible to reject by vibration all tubes rejected by the tapper.

The 32 type 6AC7 tubes had been subjected to vibration for five runs, and to five runs on the tapper peak and average channels. Comparing now the rejections by the two methods, the vibration test had rejected one tube on all five runs, one tube for only four runs and two tubes for only three runs, for a total of four tubes rejected three or more times. The corresponding figures for the pendulum tapper system were: Three tubes rejected on all five runs, two tubes rejected only four times and three tubes rejected only three times, for a total of eight tubes rejected three or more times. On a tube-for-tube basis, the tapper had rejected the very same tubes rejected by vibration, plus four additional tubes.

Restricting our attention now to the tubes consistently rejected by the two methods, the single tube rejected five out of five times by vibration was consistently rejected by the tapper as well. In addition, the tapper rejected two more tubes. One of these two had been rejected only three out of five times by vibration, but five out of five by the tapper. The other tube, which was also rejected every time by the tapper, fell well below the rejection level in all five vibration runs.

This last tube verified the last of the predictions that had been made: namely, that vibration cannot reject all tubes rejected by tapping. In order to accomplish this, nearly all of the tubes under test would have had to be rejected by vibration. This confirms the a priori reasoning that, while a strong relationship exists between the two tests, the tapper test (even when degraded with a 100-cycle filter) reveals more information about tube defects than does the vibration test.

The explanation for the behavior of this particular tube lies in the nature of the tapper excitation. In attempting to degrade the tapper system, so as to place it on a comparable basis with vibration, only the electrical output was limited in frequency range by the filter. The lack of a mechanical filter for degrading the wide-band mechanical impulse serves to explain why additional tubes are failed by tap and not by vibration, even though the electrical filter has supposedly equalized the two.

The tube output when tapped is indicative of the disturbance caused by every frequency up to the 100-cycle limit of the filter and in addition, may contain information concerning higher frequency mechanical resonances (since the tube elements are actually vibrating at these frequencies) even though they are explicitly absent from the noise measurement because of the filter. For example, electrical and mechanical nonlinearities may yield difference frequencies which fall under 100 cycles. Another possibility is that a tube may have a resonance

not far above 100 cycles. It will be excited by the tapper, and the resultant electrical output may get through the indicator system, although attenuated.

All of this vibration experimentation has led to the inescapable conclusion that impulse excitation (as supplied by the Pendulum Tapper) may be substituted for low-frequency vibration testing, with a real gain in information obtained. This additional data comes from the wide-frequency band mechanical excitation and the simultaneity of application of all of these frequencies. The other correlative advantages are the same as those enumerated under the conclusions from the resonance investigation, namely: cheaper equipment, simpler to use; increased repeatability, reproducibility and objectivity; nondestructive, more rapid testing; and simple provision for changing the frequency band under consideration.

GENERALIZATIONS

When one reflects briefly on the diversity of vehicles in which any given type of electron tube is employed, the different natural elements through, on, or under which they travel, and even further to the various locations aboard each of these vehicles, each location providing a different mechanical environment, the prospect of attaining a proper simulation of field environment in the Laboratory is not very promising.

In this respect, tubes differ from the case of large pieces of specialized equipment, where the intended locations and environments may be determined beforehand. It should also be kept in mind that the possible permutations and combinations of the above variables are further extended by the fact that two distinct classes of excitation may be present, either separately or in combination: namely vibrational, or steady-state, and transient, or impulse-type excitations. The desirability of representing these very complex field conditions with a single, simple mechanical evaluation suggests the employment of constant "g" excitation, extending from dc to as high in the spectrum as is feasible.

The unit impulse, if it were physically realizable, would be a very attractive method for the excitation of electron tubes, in the light of this consideration. By unit impulse is meant, of course, an infinite force applied to the tube for infinitely short duration. Mathematically, it can be shown that such an idealized excitation provides an infinite frequency spectrum of uniform amplitude. Furthermore, unlike swept-frequency techniques, the excitation is applied at all these frequencies simultaneously, the importance of which has already been shown. It is just this characteristic of simultaneity which enables impulse excitation to simulate the effects of vibration over a broad range of frequencies, in addition to simulating impulses, while the converse is not true. Other general advantages of impulse excitation are its speed and simplicity of equipment.

No claim is made that the pendulum tapper provides a unit impulse; however, the impulse it delivers possesses the important property, in common with the unit impulse, of simultaneity. Furthermore, the spectrum of the acceleration introduced by the impulse into electron tubes is flat from dc to well up in the audioband, and contains energy over a broad enough frequency band to excite most of the important structural resonances.

It is our belief that these two requirements should be fulfilled by any excitation intended to represent, or simulate, generalized field environments. Swept-frequency methods can, at least in theory, meet the wide-frequency band requirements. The phase "in theory" is used because, for the present, the equipment to do this is available only for very light-weight loads involving a single miniature tube, although a quite large and expensive equipment has very recently appeared on the market which extends the frequency to 5000 cycles for heavier loads. The limitations of lack of simultaneity, however, and the time requirements for testing still leave much to be desired in this method.

Another related investigation nearing completion at the Material Laboratory concerns a study of the phenomenon of fatigue of electron tubes. Since the concept of fatigue failure requires an excitation which is nondestructive over a short interval of time, but effective over a period of many hours, in order to conserve time it is mandatory that many tubes be evaluated simultaneously. Keeping in mind the dual requirements of broad-frequency band of excitation and simultaneity of application of all frequencies, our first approach to a solution was the mechanical generation of impulses, which was later obtained on a modified electrodynamic shaker. It was found that, for normal electron-tube structures, random noise can also be considered as meeting these criteria.

These three methods have all yielded field-type fatigue failures in our laboratory. However, it is to be kept in mind that the equipment

required is very bulky and expensive, and is only justified for fatigue evaluations. For all test procedures which require determination of the mechanical properties of a tube at a given time (such as impulse noise and microphonism, resonance, and vibration evaluations) wherein it is feasible to test one tube at a time, we believe a simple impulse approach provides the best answer at the present time.

SUMMARY AND CONCLUSION

In brief summation, the superiority of impulse excitation over low-frequency, steady-state, and swept-frequency vibration has been demonstrated by certain electromechanical tests for electron tubes. The Material Laboratory system has provided all the advantages implicit in transient testing by introducing a sharp impulse directly into the tube envelope. These advantages are broad-frequency band and simultaneity of excitation, objectivity of indication, repeatability and reproducibility, coupled with speed and simplicity of equipment.

From a broader point of view, perhaps the most important factor in favor of impulse testing is its ability to simulate the most general type of excitation experienced in the field. It is felt that the superiority of the method warrants considering extension to the mechanical testing of other components and equipment.

ACKNOWLEDGEMENT

It is a pleasure to acknowledge the valuable contributions to this work of L. N. Heynick and S. Winkler (7), as well as the aid provided by other members of the Electron Physics Section of the Material Laboratory.

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DISCUSSION

I. Vigness, NRL: Could you or Mr. Stinchfield make a comparison of the differences or likenesses of your two methods of excitation?

Wohl: I think it might be good to do that. I believe that the results obtained will be quite similar since both methods give you a degree of simultaneity of excitation at all frequencies over a very broad band. To some extent there is not simultaneity in the case of random noise because truly instantaneous frequencies probably are not present. However, we have done a good deal of work on this question at the laboratory and we have determined that, due to the mechanical structure of the tubes, before the transients generated at a frequency at a given moment have time to die down appreciably, the rest of the spectrum of the random noise comes into the picture. So in effect it is very close to simultaneity.

Speaking of the systems only, I would like to point out for impulse over the random excitation, the simplicity of the equipment, repeatability and reproducibility. You have a high degree of reproducibility with no necessity for calibration by accelerometer, etc., because of the nature of the instrument. You have objectivity of decision by means of a very simple thyratron circuit; and finally there is the difference of expense between the two systems.

As far as I am concerned I believe that random noise has its greatest application when you are going in for fatigue excitation rather than when you are trying to get a picture of the tube output at a given instant. With random-noise excitation you get an averaging effect over time for one thing, a good measure of repeatability, and finally you can put a large number of tubes in the machine simultaneously.

Bondley: Do you have anything to add to this, Mr. Stinchfield?

Stinchfield, Diamond Ordnance Fuze Laboratories: Yes. I believe that the impulse-excitation method is a good one and certainly an improvement over the old 40- or 60-cycle generator. It is a big step forward because it excites all frequencies simultaneously and gives you wide spectra, and I should emphasize that in our work we found we had to have a lot of high frequencies to get a really good test.

Now this impulse excitation is relatively simple. You have got a lot of different kinds of tubes to test. You must have relatively simple apparatus for testing. It has some limitations certainly. In our work, as Mr. Dorrell indicated, we used hand tapping in order to get some of these results, but it wasn't a satisfactory test certainly, and in our actual tests, we have a steady vibration. We believe that white noise is the best way of doing it. In fact, we find it a good way of making the tests. The impulse excitation would not tap the tube at the right time.

The white noise excitation is there all of the time. When the filament drifts through a range f frequency which must coincide or approach another resonant element in the tube to make noise in the passband of our circuit, you have to tap it at just the right moment. With white noise the excitation is there all of the time.

So we couldn't get satisfactory results by impulse testing, but that doesn't say it isn't a good method for general testing.

Wohl: May I add one point in comparison of these two systems? We have applied the tapper device to receiving tubes and good-sized transmitting tubes. Equipment today is not available to give random-noise excitation to anything except fairly light loads and that, of course, I think, is another advantage of the impulse excitation.

* * *

MEASUREMENT AND EVALUATION OF ACCELERATION IMPARTED TO ELECTRON TUBES IN ELECTRONIC EQUIPMENT

O. A. Biamonte, SCEL, and A. W. Orlacchio, Gulton Mfg. Co.

A new test system has been developed which measures acceleration at tube socket locations on military equipment. Electron tubes are replaced by tri-directional accelerometers approximately their size, shape, and weight. Acceleration levels are automatically recorded.

INTRODUCTION

Historical and Purpose

Several years ago, the Signal Corps Engineering Laboratories embarked on a program of Electron Tube Reliability. One phase of this program, included as part of the technical requirements for equipment designers, a chart, known as the 516 forms. These charts are entitled, "A Summary of Electron Tube Operating Conditions." The summary chart is a complete tabulation of all operating characteristics and potentials measured at each socket location for each electron tube used in the specific equipment. Among other things, it requires that the environmental condition—that is—the peak acceleration at each socket location be tabulated for the lateral and longitudinal planes when the equipment is subjected to the "black box" environment specification.

It was felt that data regarding the environment at socket locations resulting from transmissibility would possibly reveal:

- Inadequate tube specifications,
- Poor mechanical design of chassis,
- A combination of both thereby masking the real electron tube problems.

The use of the 516 forms went into effect at SCEL early in 1953 and it has been found to be a useful tool with regard to detecting faulty circuits and misapplication of electron tubes prior to large-scale production of the specific equipments. Unfortunately, the requirement for the accumulation of data for the environmental columns was waived because of the lack of the proper measuring equipment. For this reason, the equipment about to be described was developed under the sponsorship of the Signal Corps.

General Description

In general the equipment consists of sensing elements, an automatic sampling selector, multichannel amplifiers, and recorders. Provisions are made for monitoring the output of the accelerometers directly for analysis purposes. Before going into further detail of the equipment, and to be in line with the theme of this symposium, I should like to say that we in the Electron Devices Division of SCEL utilize a package tester as a Laboratory simulator for vehicular or transportation environments. The environment can best be described by visualizing a jeep jogging along a rough terrain. It is a random shock and vibration with frequency components up to 2,000 cps.

The package tester (Figure 1) depicts the entire test system to be described. The test system can be seen to contain:

Figure 1 - Test system used on SCEL program - electron tube reliability

- The sensing elements,
- The automatic sampling selector, which contains the cathode followers for matching of impedances,
- The electronic instrumentation which includes the multichannel amplifiers, filter networks, recorders and power supplies.

Figure 2 shows the test system used to record the amplitude of vibration at socket locations resulting from a single-frequency vibration specification requirement. Incidentally, should it be desired to measure the environment at socket locations resulting from variable-frequency vibration specification, all that need be done is to synchronize the sampling selector time with the variable-frequency cycle time.

The test system as designed is ideal for making socket location measurement by using the vibration equipment excited by random, white-noise specification requirements.

DETAILS OF EQUIPMENT

Over-all Design

A block diagram of the test system is shown in Figure 3. The diagram shows the accelerometers with their outputs connected to the sampling selector. The signals then go from the sampling selector to the filter network and then to the lateral and longitudinal amplifier channel, which include peak detectors, and then to the recorders.

The test system is capable of recording the outputs of 80 accelerometers each of which have two outputs recorded. The total time for recording the 160 output is ten minutes.

Accelerometers

At the time of the conception of the new test set, the Signal Corps requirements for the sensing element included omnidirectional accelerometers simulating electron tubes in form and weight. During the progress of the development, the omnidirection approach for the sensing element was abandoned and replaced by utilizing three mutually perpendicular seismic elements in each accelerometer housing and each of those has its maximum sensitivity in one of the x, y, or z direction of a cartesian coordinate system.

The z axis in this case is along the longitudinal axis of the tube-like accelerometer. The x and y axes are located in lateral plane of the tube-like accelerometer. The accelerometers

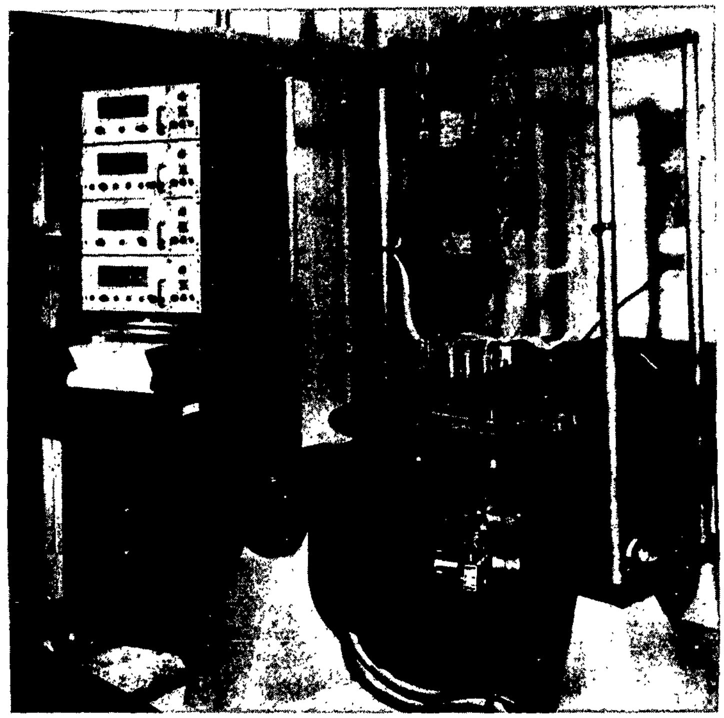

Figure 2 - Environmental measurements at socket location on electronic equipment

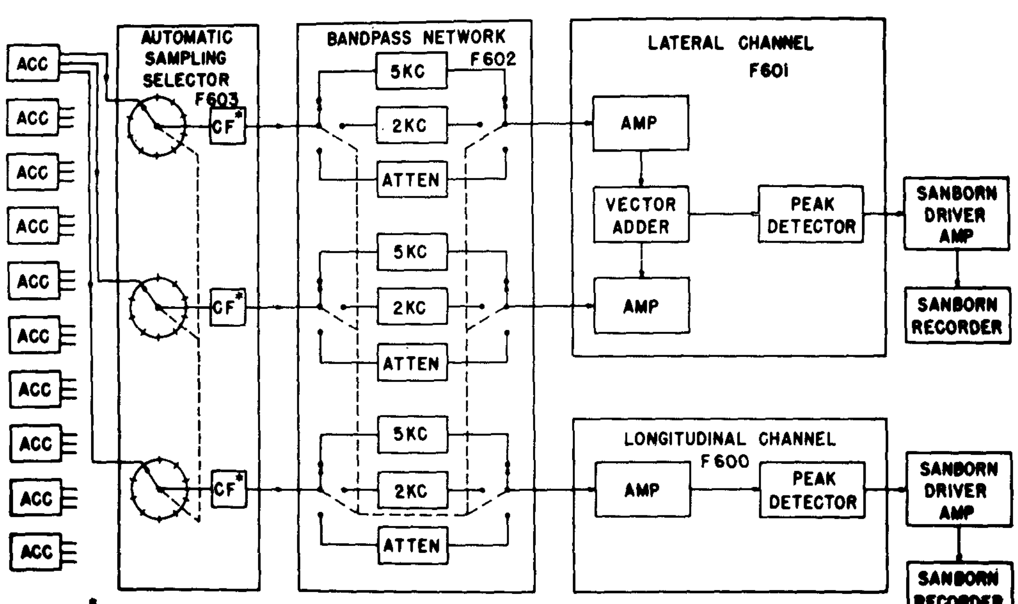

Figure 3 - Block diagram of test system

are designed to conform to electron tube types, T-2x3, T-3, T-5-1/2, T-6-1/2, and T-9's. There are two general types of accelerometers, those having their outputs coupled to coaxial connectors, and those having their outputs coupled through the pins of the simulated tube accelerometers. The two groups of five types of accelerometers were selected for adaptability to all types of chassis design. Figure 4 shows the two groups of various types of accelerometers.

Accelerometer Characteristics and Padding Condensers

The accelerometers are individually calibrated from 20 to 1,000 cps with an accuracy of 5 percent. However, since it is not feasible in production to control equal sensitivity for each seismic element, appropriate padding condensers are supplied, which when properly matched, bring the normal sensitivity of each element down to 2 m u/g. A coding system is employed wherein the appropriate numbered padding condenser is matched to the similar number engraved on the x, y, or z output of each accelerometer. The electromechanical characteristics of the accelerometers are shown in Table 1.

TABLE 1
Electromechanical Characteristics of Accelerometers

Accelerometer		Equivalent Electron Tube	
T9	43.5 gms.	T9	35 gms.
T9C	44 gms.		
T6-1/2	16 gms.	T6-1/2	10 gms.
T6-1/2C	16.5 gms.		
T5-1/2	13.2 gms.	T5-1/2	9 gms.
T5-1/2C	13.6 gms.		
T3	4.3 gms.	T3	3.5 gms.
T3C	5 gms. w/o cable		
T2x3	4.1 gms.	T2x3	3.5 gms.
T2x3C	5.3 gms. w/o cable		

Sensitivity of all elements: 2 mv/g (with appropriate shunt capacitors)
Resonant frequency of all elements: 15,000 cps.
Capacitance: greater than 1,000 mmfd
Useful frequency range: 3 - 5,000 cps
Acceleration range: 1 - 200 g
Max. lateral sensitivity: less than 10 percent
Armatures: 3 mutually perpendicular outputs in one housing
Seismic movement: bender type of elements

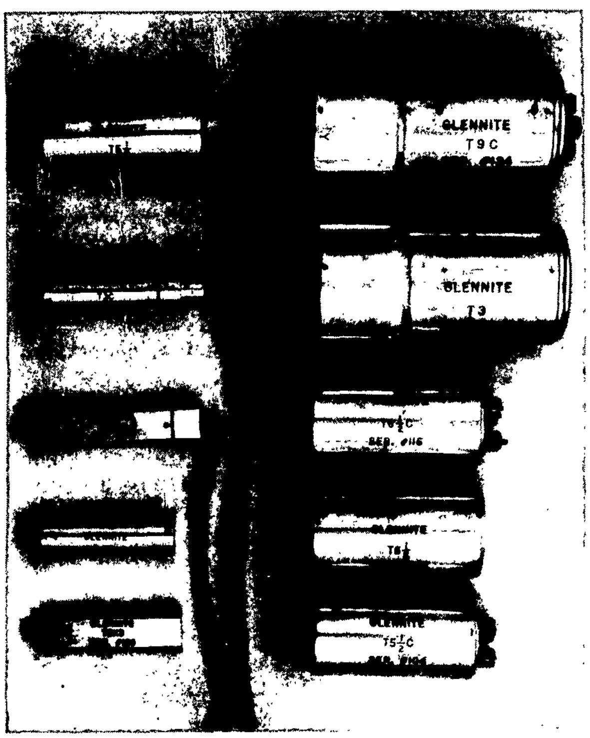

Figure 4 - Types of accelerometers used in test set

Recording System

The recording instrumentation consists of the automatic-selector filters and attenuators, lateral- and longitudinal-amplifier channels, and the peak recorders.

The automatic-sampling selector has two main functions; first, the successive selection of inputs; and second, for the necessary impedance matching between the accelerometers and the following instrumentation.

The selector consists of an input panel which has two connectors for each seismic element input or a total of six connectors for each accelerometer. The two pair of three inputs for each accelerometer are marked x, y, and z and x_1, y_1, and z_1 respectively; these are connected in parallel internally. The x, y and z connectors are for the cabling from one accelerometer and the x_1, y_1, and z_1 connectors are for the corresponding padding condensers previously mentioned. In the automatic cycling arrangement, the switch has six 10-position sections which are grouped in pairs. In each pair there is one section which connects the outputs of one accelerometer at a time to the cathode followers and grounds all other connections. The switch is driven automatically and switches each minute by using a ledex solenoid which is progressively stepped by a cam switch.

The filter networks consist of three units, one for the longitudinal input and two for the lateral input. Each unit has a 2-kc filter, a 5-kc filter, and a 6-db attenuator. Anyone of these filters or attenuators can be switched into the circuit manually.

The main components of the longitudinal channel are an amplifier and a peak detector. In the amplifier it is possible to change the gain by use of a voltage divider. This permits the selection of a unit to have full-scale sensitivity of either 2, 5, 20, 50, or 100 g. Following these amplifiers there is a double diode which operates as a peak detector—one indicating positive peaks, the other indicating negative peaks. The magnitude of all peaks are recorded.

The lateral channel has fundamentally the same operation as the longitudinal channel. However, following the amplifier, which is similar to the one used in the longitudinal channel, there is a vector adder, wherein the two incoming signals, that is the x and y signals from the accelerometers, are fed into phase splitters. The four output signals from this circuitry are then modulated, each in a separate modulator tube, with a three-megacycle carrier. The outputs of the four modulators are added across the common impedance with proper phasing adjusted in a three-section delay line, each shifting the signal by 90 degrees. The final output modulator is amplified and fed to the peak detector circuit.

The longitudinal and lateral channels include a matching stage for the driver amplifier and recorder which are standard Sanborn equipments.

Provision is made for monitoring individual filtered outputs of the seismic elements of any accelerometer. This permits the gathering of data, which can be put on tap, and when processed can give a frequency-versus-amplitude analysis at a specific socket location.

CONCLUSION

The two important features in the development of the system described have been the design of subminiature accelerometers containing three seismic elements and the development of equipment to add two orthogonal signals. Throughout this development it has been our aim to furnish a test system capable of accurate and rapid measurements and yet provide these measurements without the use of instrumentation specialists for actual operation.

This equipment has been under evaluation at SCEL for several months. While it is not yet used in the field or by equipment manufacturers, it promises to be a useful tool for the evaluation of shock and vibration problems regarding electron tubes and electronic chassis design.

* * *

SIGNIFICANCE OF VIBRATION AGING REVIEWED AND ANALYZED

R. S. Whitlock, Wright Air Development Center

Certain experimental results obtained at WADC and by other investigators, concerning the effects of vibration aging, are presented and reviewed. The general experience indicates, with few exceptions, that the user obtains no discernible benefit from post-manufacturing tube treatments such as aging.

Experimental results obtained by the Wright Air Development Center and by others are discussed in this paper. In particular, attention is directed to results which are indicative of the effects that can be expected from vibration when applied as a part of electron-tube aging procedures. By aging procedures I mean the various processes which involve the operation of tubes under some defined set of conditions for some arbitrarily chosen period.

Experience shows that tubes taken from stock and put into service will exhibit a number of early failures. Many times the environment in which tubes operate is such that mechanical excitation is present. Because the rigor of production quality control varies across the industry, tubes with poor mechanical properties find their way into stock. These are the tubes which frequently exhibit early-life field failures.

It has been assumed, by some, that if tubes were mechanically vibrated under cycled electrical operating conditions prior to use, these potential failures would be disclosed, and if failed tubes are eliminated, the residue would contain only reliable long-life tubes. It has been further assumed that this will bring a stabilizing effect on tube characteristics. The validity of these assumptions is certainly open to question and the assumptions are suspected of over-optimism. Such procedures when applied by the tube manufacturer are a necessary and legitimate part of the manufacturing process. When applied by the tube user, subsequent to his receipt of incoming material, they are open to certain objections.

To forestall a possible misconception, I would like to assure you that those of us who are, in one way or another, responsible for electron tubes in the Air Force are not fundamentally opposed to special tube treatments. We do feel that it is necessary to know, as completely as possible, just what we are doing. It is for this reason, since we have seen no adequate supporting evidence, that we have discouraged such practices, and must continue this opposition until admissible evidence is adduced to show their worth. To us this means data from controlled experiments, sufficient to show: first, that a benefit followed the treatment and second, that statistically there is little likelihood that the benefit was due to chance.

When one is under pressures to produce results rather than to verify causes, and this is more often than not the case, there is a tendency to assign cause and effect relationships in cases where the most that could safely be said is—it appears that an association exists. That the association may be entirely fortuitous frequently is overlooked. We are in trouble, we do something (whether by intent or accident) and having done it, we observe that a condition favorable to our purpose prevails. It is possible that this thing we did will be adopted, without further investigation, as standard operating procedure. In what follows, I have used such terms as aging procedures, burn-in, and stabilization period interchangeably.

In 1951 to 1953 the Air Force was engaged in just such an enterprise. We were issuing tubes from stock, giving them a test, burn-in, retest treatment and installing this supposedly improved material in equipment sockets. By 1953 we were questioning the wisdom of this procedure, questioning what, if any, benefits were resulting from the practice. What were we buying?

To answer these questions a controlled experiment was designed, equipment was installed at two Air Force activities: Gentile AFD, Dayton, Ohio, and Kelly AFB, San Antonio, Texas. Over approximately a six-month period a total of 35,000 tubes representing seven types were processed between the two activities. The aim of the experiment was to evaluate the effect of a particular treatment on material which had been delivered by the supplier. There was no intention of evaluating the material as such. The experiment was designed accordingly. The electrical conditions were the test conditions of the applicable specification.

The conditions of vibration imposed on certain test groups were: mechanical excitation frequency 30 cps, acceleration level between 1.5 and 2 g, and rack time 48 hours. Operating voltages were cycled 50 minutes on 10 minutes off.

Tubes of each type were selected to represent each manufacturer and to give a good cross section of production dates. Tubes were in every case drawn from stock in unopened carton lots of single-date code material, to secure homogeneous material. Each carton of 200 tubes was broken into four test groups. These were treated as follows:

All groups but one were electrically tested to the requirements of the applicable specification, the rejects from each group were discarded and replaced by acceptable units from the same carton. One tested group was placed in shelf storage, one was burned without vibration, two were burned with vibration, one of these being the group which received no initial electrical test. At the end of the processing period all tubes were electrically tested to the specification. The resulting data were analyzed. Conclusions drawn from the results of the analysis were:

1. No significant differences exist among double testing, burning, and burning with reference to "detriments," i.e., shorts, opens, and interelectrode leakage in electron tubes.

2. A small but statistically significant difference exists between double-testing and burning or between double-testing and burning with vibration as to "characteristic" faults (plate current, screen current, emission, and transconductance).

3. No significant difference exists in any case between burning and burning with vibration.

4. For the entire test, involving 35,000 tubes, there is no significant difference between out-of-the-box testing and testing after burning with vibration. In the individual reject categories, this comparison reveals a difference in I_b balance rejects, but this is small compared to the over-all percentage.

The experience of others may now be reviewed. The Lansdale Tube Co. examined its data from the manufacture of 6100 CT tubes produced in the periods 15 November to 14 December 1951 and 15 May to 13 June 1952. These tubes were given two successive 46-hour, cycled-voltage stabilization periods. During the second of these periods, the tubes were vibrated at 30 cps 2 g. The tubes were tested for opens and shorts. A far greater number of failures was found after the first 46-hour stabilization than was found after the second vibration, period. The investigators concluded that vibration during stabilization had little or no value.

The International Business Machines Corp. examined 23,314 tubes of various types. They constituted a sample from manufacturers production lots totaling 303,152 tubes (before any defectives were eliminated). A vibration stabilization procedure, consisting of 30 cps, 1 g vibration for 46 hours, during which time the operating voltages on the tubes were cycled, was applied by IBM. The results were compared with the tube manufacturers pre-delivery data. It should be observed that this was an uncontrolled test in that there was no knowledge of the correlation existing between the IBM test methods and those employed by the tube manufacturer. There was also an intermediate shipping and handling treatment involved. This work is reported in some detail in Technical Report NE-091105 dated 10 March 1954, prepared by the Material Laboratory, New York Naval Shipyard, Brooklyn 1, N. Y. The conclusion drawn was that the contribution of vibration to the weeding out of defective tubes was negligible.

In a Hytron Radio and Electronics Co. unpublished report by E. K. Wimpy and E. P. Laffie, dated 17 March 1953 the following appears:

"The Quality Control Department at CBS-Hytron, in conjunction with the Engineering Department, designed a series of experiments on tube types 6099/CT and 6AK5, manufactured at our Salem, Massachusetts Plant. The purpose of these experiments was to determine if we could detect any practical difference in the number of 'inoperable' tubes detected when they were subjected to 'burn-in', or simply to repeated tests without 'burn-in'.

"We have concluded from the results of our experiments, that there is no practical difference in the percentage of 'inoperable' tubes for types 6AK5 and 6099/CT, found by testing—burn-in—testing, etc., or just by testing—placing on shelf—testing, etc."

On 12 September 1955 the Eclipse Pioneer Division of Bendix Aviation Corp., Teterboro, N. J., wrote to the Electron Tube Subcommittee of AIA, in part as follows:

"The following is an abstract of a report on the effects of vibration aging of certain vacuum tubes. The data for the report was obtained by Bendix Red Bank.

"1. Samples Investigated:

A sample of 600 of each of seven receiving types was taken at random from stock

5654	6X4	12AT7
5670	5903	6AQ5
TD-9 (Bendix Regulator)		

One-half of each sample was set aside as a control, the other half was subjected to a 46-hour aging, with power cycling and vibration at 1G.

"2. Experimental Procedure:

a. Test to MIL-E-1B Standard (all tubes).

b. Vibration age for 46 hours (experimental lot).

c. Retest all tubes.

d. Store all tubes 24 hours.

e. Retest all tubes.

f. Vibrate all survivors of two types (5654 and 6AQ5) at 2.5G for 46 hours with cycled operating voltage.

g. Test tubes processed according to (f) above.

"In general, the conclusions from the statistical data taken are:

1. No significance in reliability can be attributed to the vibration aging.

2. The significant improvement can be attributed to the effects of repeated testing."

When the equipment manufacturer undertakes aging procedures his objectives are without exception laudable and not open to criticism. He is working to attain very desirable end-results which are: first, improved electrical stability of tubes and second, the removal of potential early life tube failures. The attainment of these results is certainly suspect and the procedure is open to the following objections, each of which must be refuted.

1. It can easily obscure or disguise the real source of difficulty. It may thus become a "catch-all" for wishful attempts to eliminate many non-related problems.

2. Because of the elaborate extremes to which it can be carried, it easily becomes an end in itself and may not be employed as a means of predicting equipment reliability. (It is separated from its only useful relationship—to equipment reliability.)

3. It is not a suitable adjunct to a production program because of its high cost and time consuming nature.

4. It leads an equipment manufacturer into a false feeling of security because:

   a. He has no assurance that the tubes after processing, and in any given interval of operating time, will have either as low an inoperative failure rate or will be as stable electrically as was observed during the processing period, or indeed as might have been observed prior to processing.

   b. He can easily damage the tubes electrically during the processing operation. This also is another "handling" procedure that has been shown by Aeronautical Radio, Inc. to be detrimental from the standpoint of mechanical quality.

In view of the evidence which has been brought forward to date, I would recommend, and this is a purely personal recommendation, that:

1. Tube processing should be performed only by the tube manufacturer. In addition to its questionable value when performed by others it leads to tube selection in the field and a consequent military supply problem.

2. From the equipment performance requirements, the contingent tube performance necessary for satisfactory equipment operation must be determined. The tolerances on tube stability that are necessary must be specified as must the tube mechanical-failure rate that can be tolerated. These factors can be measured and can be specified in tube acceptance specifications. They obviously apply throughout the expected tube life. A post-manufacturing processing procedure, as such, gives no assurance of what will happen during the normal tube life.

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